REHEATING COMPRESSED AIR

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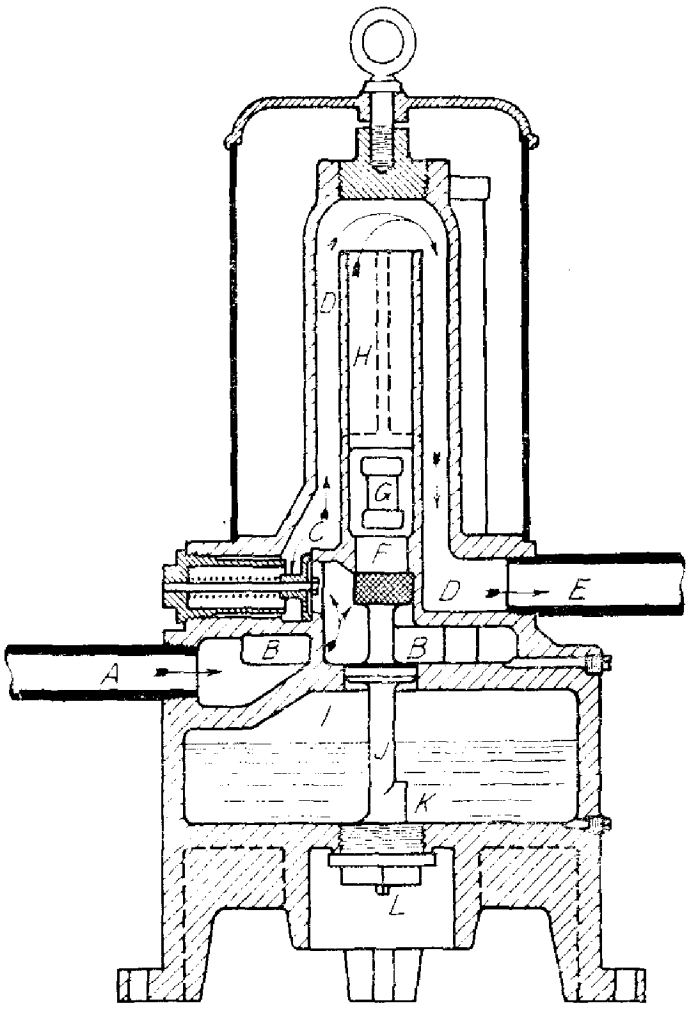

FIG. 6. SECTIONAL VIEW OF INTERNAL COMBUSTION REHEATER

Simons

CHAPTER XVI

REHEATING OF COMPRESSED AIR

122. In preceding articles it has been shown that the available energy residing in a given weight of compressed air at the end of the pipe line is considerably less than that which the same weight of compressed air could develop immediately after leaving the compressor. This is due to the fact that the volume of a given weight of air, having a given pressure, is smaller at the lower temperature which prevails at the end of the pipe line than that which it occupies when leaving the compressor at a high temperature, and to the fact that the available power residing in compressed air is dependent on volume as well as on pressure.

This has led to the introduction of a process known as "Reheating." By this process the volume of the compressed air at the terminal may be, by heating, increased so as to partly or completely make up for loss of power in transmission.

To accomplish this result there must be an expenditure of fuel. This expense, however, is very light. For the average air engine it amounts to about one-seventh of the fuel that would be originally required to compress air so that it would be in a condition to develop an equal power, it being assumed that coal is the fuel used. This is due to the fact that the average efficiency of a Corliss steam engine does not exceed 10 per cent., based on the total heat value of the fuel, whereas in reheating in a proper heater 70 per cent. of the heat value of the fuel may be utilized. The increase in efficiency resulting from reheating makes it possible to use a much smaller air compressor for performing a given amount of work.

In addition to increasing the efficiency, the reheating of compressed air also prevents the freezing of the exhaust ports of air engines which often becomes troublesome when air containing considerable moisture is exhausted at temperatures below the freezing point.

Let us assume that at the end of the pipe line we have 1 cu. ft. of air at 75 lb. gage, and at a temperature of 60° Fahr. and that we wish to double this volume by reheating.

The volume of free air which must be compressed to make, after being cooled down to 60° Fahr., 1 cu. ft. of air at 75 lb. gage, we find from the equation

$$V = V_1 \frac{P_1}{P}$$

whence $$V = 1 \times \frac{89.7}{14.7} = 6.10 \text{ cu. ft.}$$

To compress adiabatically in one stage 6.10 cu. ft. of air to 75 lb. gage and deliver it into the receiver or the pipe line requires work to the amount of

$$W_n = \frac{144\,nPV}{n-1}\left[\left(\frac{P_1}{P}\right)^{\frac{n-1}{n}} - 1\right] \text{ foot-pounds (theoretical)}$$

$$= \frac{144 \times 1.406 \times 14.7 \times 6.10}{0.406}\left[\left(\frac{89.7}{14.7}\right)^{0.29} - 1\right] = 30{,}832 \text{ ft.-lb.}$$

Adding 15 per cent. for friction we have

$$W_n = 35{,}457 \text{ ft.-lb.}$$

If the compression is to be accomplished by a steam engine cutting off at 1/4 stroke the number of cubic feet of 75 lb. steam required to do the work is found as follows:

$$W_n = 144\, P_m V_1$$

in which, W_n = work in foot-pounds = 35,457
P_m = mean effective steam pressure in pounds per square inch = 37.8
V_1 = volume of steam in cubic feet after expansion = four times the volume we wish to ascertain.

Introducing values $35{,}457 = 144 \times 37.8 V_1$

whence $$V_1 = \frac{35{,}457}{144 \times 37.8} = 6.52$$

Dividing by 4 $$\frac{6.52}{4} = 1.63$$

Adding 15 per cent. $$1.63 + \frac{1.63}{100} \times 15 = 1.87 \text{ cu. ft. of 75 lb.}$$

steam, or practically 2 cu. ft.

In other words, to compress a certain mass of air by steam pressure so as to furnish 1 cu. ft. of compressed air at 75 lb. gage and

at 60° Fahr., requires practically 2 cu. ft. of steam at 75 lb. Now, 1 cu. ft. of steam at 75 lb. weighs $\frac{1}{4.86}=0.206$ lb. (Kinealy steam engine). Total heat required to make 1 lb. of steam at 75 lb. from water having a temperature of 60° Fahr., is:

$$1179-(60-32)=1151 \text{ B.T.U.'s (Kinealy).}$$

Total heat required to make 2 cu. ft. of 75 lb. steam, is:

$$2\times1151\times0.206=474 \text{ B.T.U.'s}$$

From this it follows that the number of heat units required to produce by steam energy 1 cu. ft. of air at 75 lb. gage and at 60° Fahr. is 474 B.T.U.'s.

The temperature of 1 cu. ft. of air at the end of the pipe line is $(60+461)=521$ degrees absolute. To double the volume at constant pressure, we must double the temperature, that is, the absolute temperature of the 2 cu. ft. of air at 75 lb. gage would be 1042 degrees and the increase in temperature is

$$1042-521=521 \text{ degrees.}$$

The weight of 1 cu. ft. of air at 75 lb. gage and at 60° Fahr. is from Article 24:

$$W_2=2.7077\frac{P_2}{T_1}$$

$$W_2=2.7077\frac{75+14.7}{60+461}=0.466 \text{ lb.}$$

The specific heat of air at constant pressure is 0.2375. Therefore, to raise the temperature of 0.466 lb. of air at constant pressure by 521 degrees requires

$$0.466\times0.2375\times521=58 \text{ B.T.U.'s}$$

This shows that to produce an extra cubic foot of air at 75 lb. by a steam-driven compressor would require 474 heat units, whereas by reheating we have at an expenditure of only 58 heat units made 2 cu. ft. out of the original 1 cu. ft. of compressed air.

The additional power cost of reheating, expressed in heat units, in this case is therefore only one-eighth of that of compression by steam energy.

In practice, no attempt is made to double the volume of compressed air, after it arrives at the air engine, because at temperatures much above 300° Fahr. the lubricant in the motor is apt to charr, causing severe cutting action on the valves, rods, and stuffing boxes.

To increase the volume of the compressed air by reheating from 40 to 50 per cent. is considered quite satisfactory. Beyond that, the air is heated only sufficiently to compensate for heat loss during its passage from the heater to the air engine. To minimize this loss, the heater should be placed as near the point of use as circumstances will permit, and the pipe between the heater and the machine should be well covered.

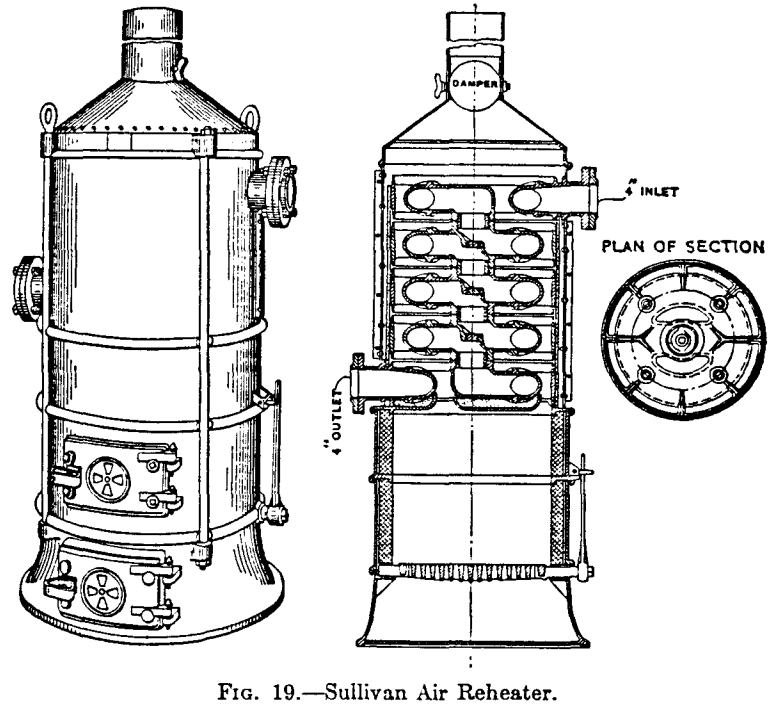

FIG. 19.—Sullivan Air Reheater.

In the following articles are illustrated and described two types of reheaters which are being used in operations where compressed air is employed for power purposes.

AIR REHEATERS

123. The Sullivan air reheater, illustrated in Fig. 19, consists of a series of hollow annular rings, forming the heating surface,

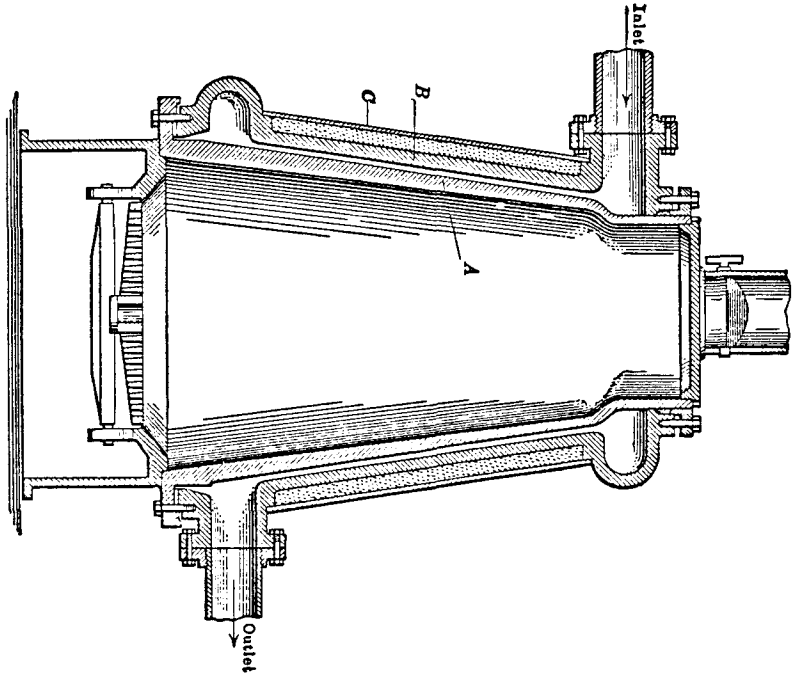

Fig. 20. Sergeant Air Reheater.

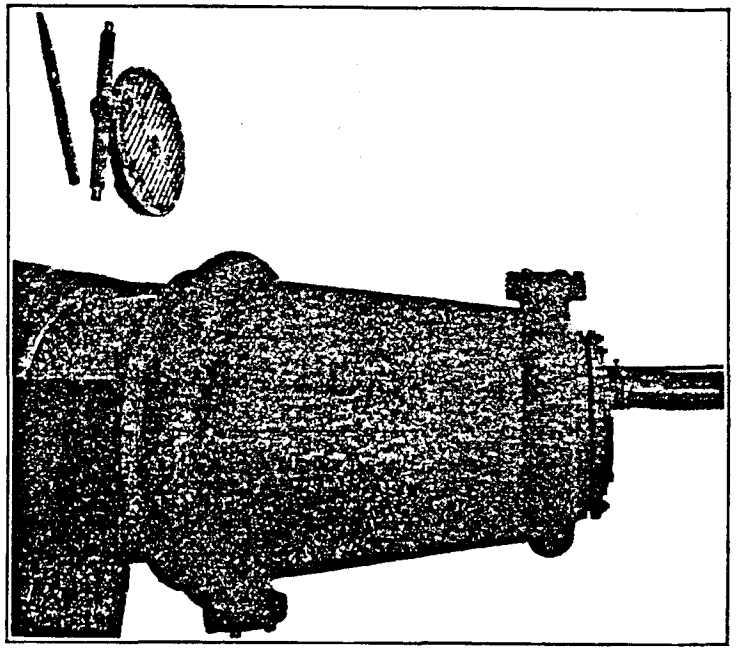

Fig. 20A.

surrounded by asbestos matting and enclosed in a sheet-steel shell. The rings and shell rest upon two cast-iron rings, lined with fire brick, forming the fire box. The latter is provided with dumping grate and doors as shown. The hot gases after circulating around the rings, escape through the hood and smoke pipe on top.

Air enters the reheater at the top and is forced to take a circuitous passage through the annular rings by means of baffle plates so that it comes in contact with the heating surface. The heated sections are designed so as to prevent leakage in the joints, due to expansion. The heated air leaves the reheater by a flanged opening in the bottom ring, similar to that by which it enters the top ring.

These reheaters are designed for burning coal, but may be adapted for oil fuel.

124. The Sergeant air reheater, made by the Ingersoll-Rand company, is shown in Fig. 20. The air enters at the top of the heater, is forced in thin sheets through the annular space between the inner shell (*A*) and outer shell (*B*) of the heater, and leaves the latter at the bottom. The increased air space between intake and discharge pipes, due to the conical shape of the castings, provides for the expansion of the air in heating. The outer shell is surrounded by a mantel (*C*) of sheet iron and the space between the latter and the shell (*B*) is packed with asbestos.

125. Other reheaters, using steam, are successfully employed for surface work. Those described are, in general, not suitable for underground work in mining operations where the smoke from coal or oil fuel is objectionable. Although a number of appliances have been tried for such work, no heater that could be used satisfactorily under all conditions has made its appearance as yet.

Chapter XV.

COMPRESSED-AIR REHEATING AND ITS WORK.

ONE of the most important economies in the use of compressed air is the saving obtained by the increased volume due to reheating. The first efforts made in this line were probably suggested by the tendency of rock drills to become so frosted in the exhaust as to interfere with their best work.

Experiments made by placing a wad of oily waste in a chambered fitting, close to the steam chest of a drill, which was found to burn freely fed by the passing air, led to trial of a miner's lamp in a small chamber, by which arrangement the products of combustion were added to the compressed air and, passing through the cylinder, modified in a great measure the intensity of the frosty exhaust.

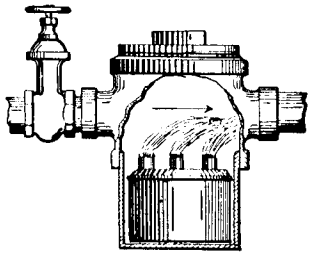

FIG. 61.—SIMPLE REHEATER.

In Fig. 61 this simple reheater is illustrated in its primitive form. The experiment clearly demonstrated the possibility of utilizing the heat and products of combustion for their full value.

Another experiment in the same line is shown in Fig. 62, in which any easily combustible fuel can be enclosed in a chamber above a wire-gauze partition in an enlarged fitting close to the air chest. An opening in the fitting, not shown in the cut, allows of igniting the combustible in contact with the wire gauze, when the combustion is kept up by the passing air and is fed by gravity from above. This method of reheating intensifies combustion and is fairly safe in mine drilling and pumping.

Another form of internal combustion reheater, patented by "Edison," is illustrated in Fig. 63, and consists of a chamber

within a chamber, between which the air flows and is heated by a fire within the internal chamber. A by-pass regulated by a valve allows enough air to pass under the grate to feed the fire. A jacketed pipe leads the products of combustion from the top of the fire-chamber to the following main air pipe, also regulated by a valve. A closure in the main intake air pipe produces a differential pressure which insures circulation through the fire-chamber. A hand-hole plate at the top fastened by a yoke and screw allows of access for feeding fuel, and a full-sized head and yoke at the bottom allow of thorough cleaning. In ordinary operation the fire can be fed and ashes blown out without interrupting the main flow of air, by operating the by-pass valves.

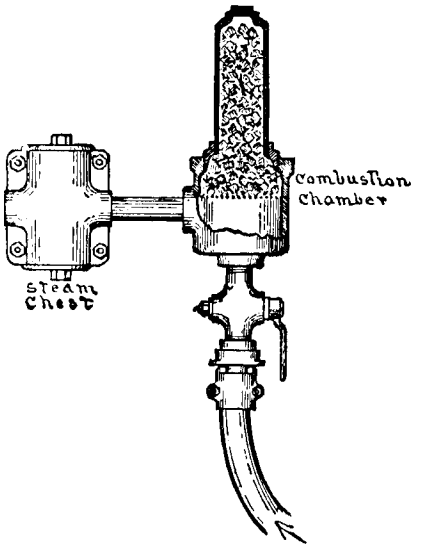

FIG. 62.—ROCK-DRILL REHEATER.

Reheaters of the class used in the Popp compressed-air system in Paris are made with pipe coils in a stove for small motors, and with cast-iron double-chambered stoves in which the products of combustion are carried to a chimney and wasted. The Sergeant reheater (Fig. 64) is a double-chambered stove in which all the compressed air passes vertically through the space between the fire-box and the outer shell. The fire is fed from the top, and can be stoked through the open grate at the bottom.

This form of reheater is in general use, and is as simple and easily managed

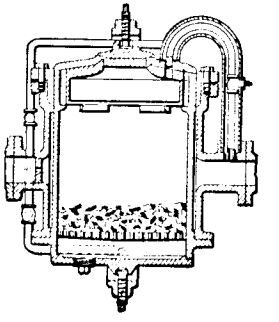

FIG. 63.—EDISON REHEATER.

as is possible under most of the conditions available for economizing the use of compressed air in motor engines.

From tests made with this heater it has been found capable of heating 340 cubic feet of free air per minute at 40 pounds pressure to 360° F., giving a gain of 35 per cent in the meas-

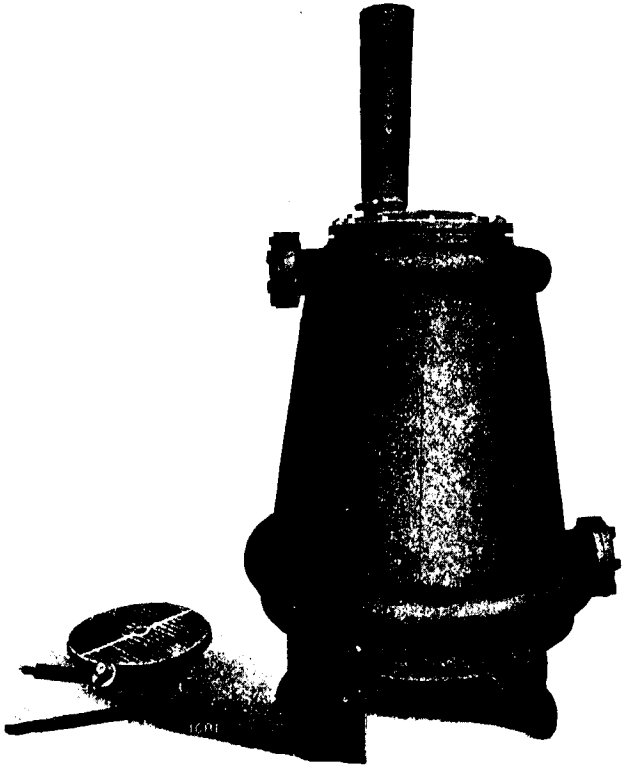

FIG. 64.—THE "SERGEANT" REHEATER.

ured amount of work done by the air after passing through the heater, compared with the same volume of air when used cold.

A heater of this size will heat less air to a higher temperature or more air to a lower temperature, than stated above; but if it should be required to heat more than 400 cubic feet of free air per minute, to get the best economy it is advisable to use the heaters in series, allowing about 400 cubic feet of free air per minute for each heater. The heater should be placed

as near as possible to the point where the air is to be used, and the outlet pipe should be as short as possible and well covered, so that the air will retain its heat.

Trials have been made in reheating compressed air by injecting steam into the air pipe near the motor, by passing the air through a steam boiler, and in the Mekarski and other compressed-air systems by passing the air through a tank charged with water at a high temperature.

Experiments have been made with a combination of steam with compressed air with an economy of 25 per cent in air volume by an expenditure of 1½ pounds of coal per horse-power

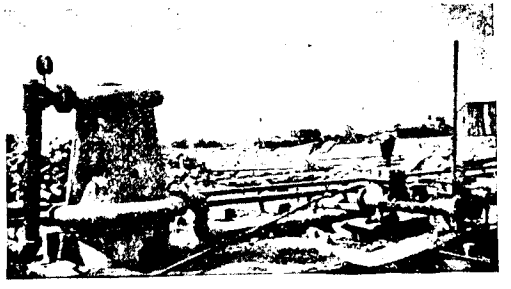

FIG. 65.—THE "SERGEANT."
Reheating the air for rock-drilling and pumping in Jerome Park Reservoir.

hour for the steam used, and was found to be equivalent to an additional horse power for each pound of coal burnt in the heater.

In consideration of the unavailability of steam except in a few locations where steam at high pressure is in use near the location of compressed-air engines, the heating of compressed air by steam for motors is of little or no practical value. Reheating by the hot-water system as used on railway cars has proved very economical.

The reheater of the Rand Drill Company, New York City, is illustrated in Figs. 66 and 67, and has a furnace lined with fire-brick and an ordinary fire and ash-pit door. The heating surfaces are composed of concentric annular spaces of gradually

increasing area, keeping the velocity of the expanding air constant. The air enters at the side of an annular chamber shown in Fig. 67, passing around the heater and upward and down-

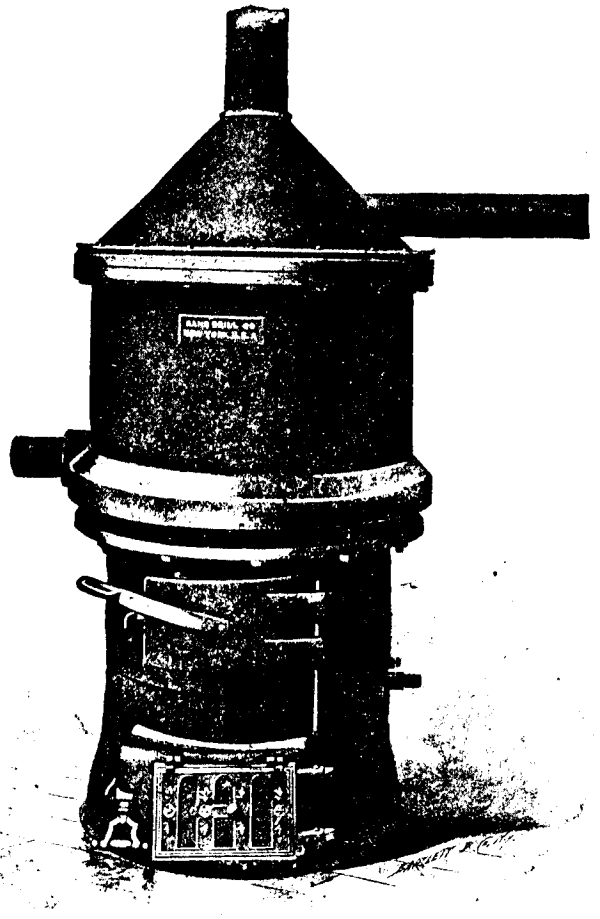

FIG. 66. THE RAND REHEATER.

ward and then upward through the thin annular spaces, making its exit at the top of the interior and hottest space.

In a test with a reheater of this type having 8½ square feet of heating surface, 530 cubic feet of free air under 60 pounds pressure were heated from 84° to 376° F. in one minute; with exhaust air from the motor, as a forced draft, the temperature

was raised to 450° F. for the same quantity of air; 300° is the most practicable temperature to operate motors and drills on account of oil lubrication; but the air temperature at the reheater may be higher to compensate for the distance of transmission.

The use of superheated water forced into tanks for car motor service has become an established system, showing the best

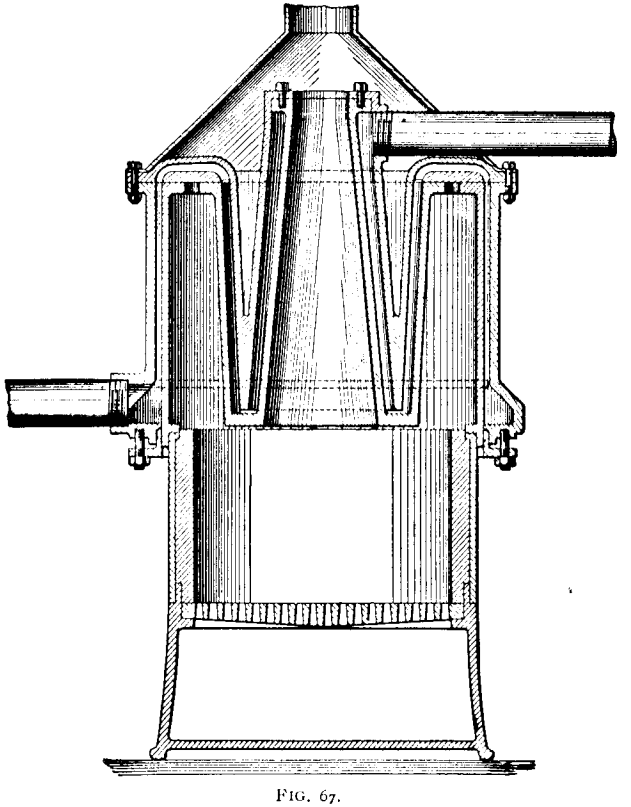

FIG. 67.

economy for this class of service, and seems to be the only available means that does not require the management of a fire on the motor car.

The hot-water reheater of the Mekarski system as used on a number of compressed railways in England, France, and Switzerland is illustrated in Fig. 68. The reheater is charged at the station with water at about 100 pounds or more pressure

at a temperature at or above 338° F., containing nearly 1,200 heat units per pound of water. In the early water reheaters of this class the air was injected from a nozzle in the bottom of the heater as shown in the cut, and thereby absorbing water vapor to saturation with but little excess of steam. In Fig. 69 is illustrated the details of this reheater as used on the Nantes, France, compressed-air tramway, in which the compressed air enters the heater at the side near the bottom, and is divided into small streams issuing from a perforated pipe and, bubbling up through the water, becomes heated to the temperature of the water, and also takes a considerable excess of hot vapor or steam, depending upon the relative pressures of the air and the pressure due to the temperature of the water.

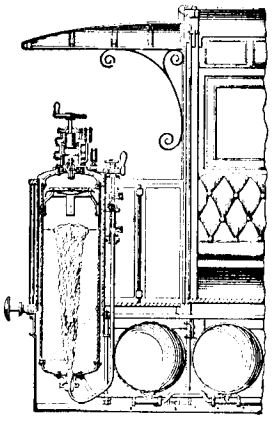

FIG. 68.—MEKARSKI REHEATER.

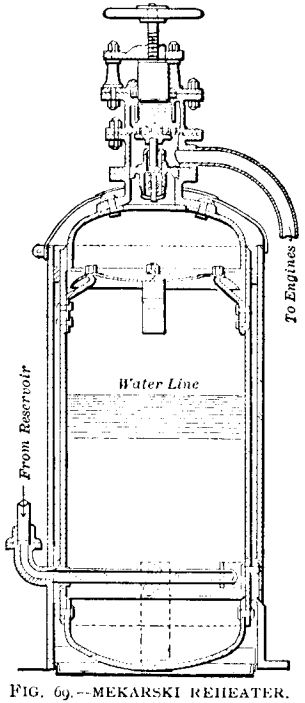

FIG. 69.—MEKARSKI REHEATER.

A diaphragm above the water line serves to prevent particles of water from escaping through the reducing valve when thrown up by the agitation of the passing air. The reducing or regulating valve is of a peculiar construction as shown in the cut. The hand-wheel when turned lowers or raises a plunger; this acts upon a liquid contained between it and a diaphragm resting upon the head of a spring valve closing against the reservoir pressure.

Just around the plunger there is an annular air space acting as an air vessel. When the plunger is depressed

into the liquid, the result is to compress the air in the air vessel to any desired extent. Then, on the air cock being opened, air bubbles through the hot water, and rises past the cone valve, which is attached to the diaphragm into the space below it, so as to press on the under-side of the diaphragm and tend to raise it; but it cannot do so until the pressure of the air below the diaphragm equals that in the annular air vessel above, and thus the pressure in the annular air vessel is automatically the measure of the pressure that will prevail in the engines. So soon as this is exceeded the diaphragm rises and closes the valve; and so soon as it falls the air in the annular air vessel re-expands and lets in more compressed air. In this way the driver can, from time to time, vary the pressure by his hand-wheel, confident that, whether the engines are running quickly or slowly, the pressure will be steadily maintained. The automatic regulating valve and the employment of the "hot-water chamber" are the distinguishing features in this particular system—the Mekarski system—of using compressed air.

The economic value of reheating compressed air in close proximity to an air motor or engine by a surface heater of the Sergeant, Rand, or Edison type is fully shown in column 2 of Table XV., which gives the increase in volume from any initial temperature to any other temperature at which the air emerges from the heater.

In a surface heater of good form the loss of fuel heat from radiation and by the smoke-pipe should be no greater than 50 per cent of the total heat value of the fuel, or, say for coal, a useful effect of 7,000 heat units per pound. This should heat $\frac{7,000}{.2375} = 29,463$ pounds of air 1° F. If to be heated from 60° to 360°, at which temperature the volume would be increased, as found in column 2, Table XV., from 13 cubic feet to 20.63 cubic feet per pound; or $\frac{13}{20.63} = 63$ per cent by reheating to an amount of 300°. Then $\frac{29,463}{300} = 94.8$ pounds heated from 60°

to 360°, and 94.8 × 13 = 1,232.4 cubic feet of initial free air, heated from 60° to 360° by 1 pound of coal. The increase in volume equals 1,232 × .63 per cent, or 776 cubic feet. Then if 10 cubic feet per minute represents 1 horse power in an air motor at any specified pressure, there should be a production of 77 horse power by reheating air to an amount of 300° by the burning of 1 pound of coal per minute, or 1.28 of a horse power per 1 pound of coal per hour, a far better result than can be anticipated from any known condition of steam power.

When the entire products of combustion are utilized there is no loss save radiation, and we can safely count on 90 per

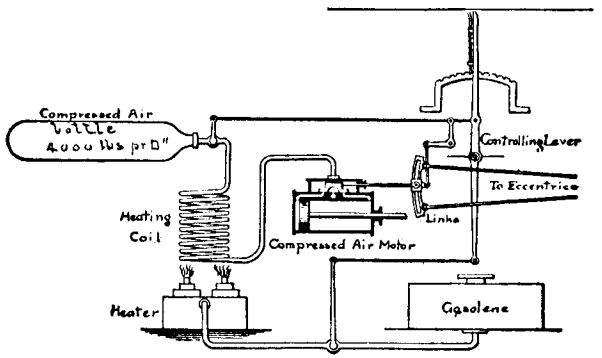

FIG. 70.—AUTOMOBILE REHEATER.

cent of the total heat units for effective work in reheating compressed air for power. Thus by the internal combustion system the saving of 2.4 horse power per pound of coal per hour may be accomplished.

The method of reheating compressed air for automobile motors is shown in Fig. 70. The air stored at high pressure issues through a copper coil at reduced pressure controlled by link-valve gear, and reheated in its passage to the motor by gasoline or kerosene burners.

Small storage-bottles of steel are made to hold 260 times their volume, or about 4,000 pounds pressure per square inch.

THE CALORIC OR HOT-AIR ENGINE.

The expansion and contraction of air by the absorption and elimination of its element of heat give to air a power for work which has been utilized to a small extent for motive power during the past century. The open-cycle system of its application in the early motor engines did not prove satisfactory or efficient.

The most satisfactory and efficient system has been derived from Carnot's suggestion of the closed cycle of heat transfer in which the pressure element of air is kept within the motor, while the heat element is generated from the outside, transmitted through the enclosed air for work, and eliminated at the cold end of the cycle by a cooling medium; and then the air is returned to the heat-imparting chamber by the alternating of two pistons.

This is the type of the action of the Ericsson pumping engine with tandem pistons in a single cylinder, and the Rider two-cylinder hot-air engine. Other hot-air engines are of the Roper type in which the heat products of combustion from an internal furnace are absorbed in or mixed with the air in its open-cycle progress through the motor, the furnace being fed with air from a pump driven by the motor. The hot air and gases are exhausted from the cylinder at the close of each power stroke.

Some trouble has been found in this class of hot-air motors from the ashes lodging in the working parts, and so clogging and wearing the surfaces.

The rapid wear of working surfaces of valves and cylinder, and the difficulties in properly lubricating caused by the intense heat and ashes, have retarded their general use, apart from their bulky proportions.

The Stirling hot-air engine, used in England and on the continent from 1816 and further improved about 1827, operated

on the closed-cycle system with a regenerator, using the air at constant initial volume with pressures due to change of temperature and intensified by the capacity of the regenerator for

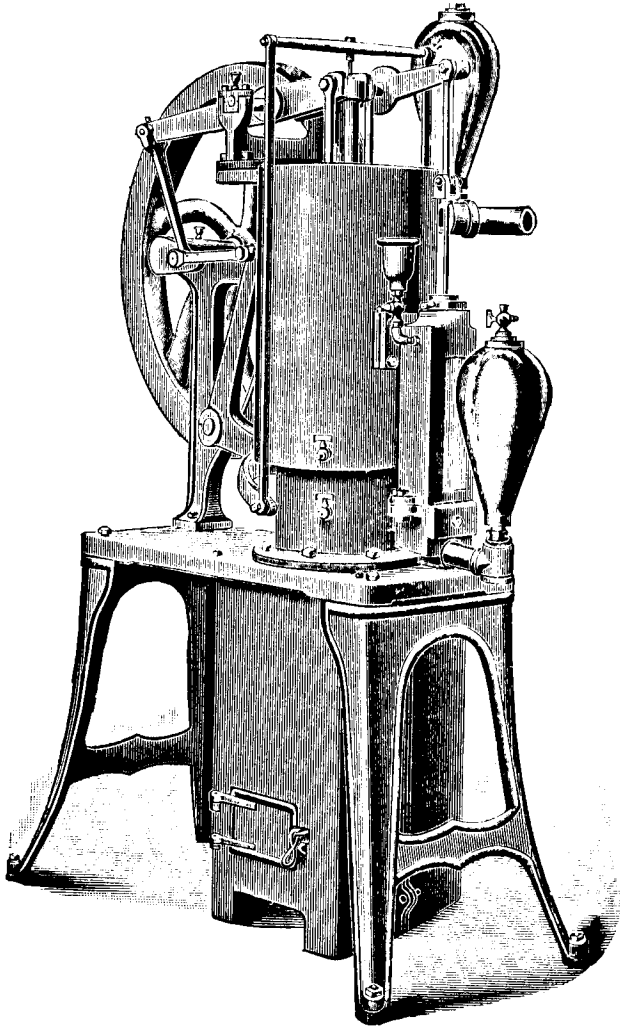

FIG. 71.—THE ERICSSON PUMPING ENGINE WITH COAL-FIRE FURNACE.

the absorption and elimination of heat from and to the air as it passed between the heating and cooling surfaces in the cycle. This engine required two cylinders, one of which was the power and cooling cylinder, and the other was the heating cyl-

inder containing the transfer piston. The modern Rider hot-air engine is an improved type of the Stirling engine.

The Ericsson modern type of pumping engine, as made by the Rider-Ericsson Engine Company, New York City, has the power piston and transfer piston working tandem in the same

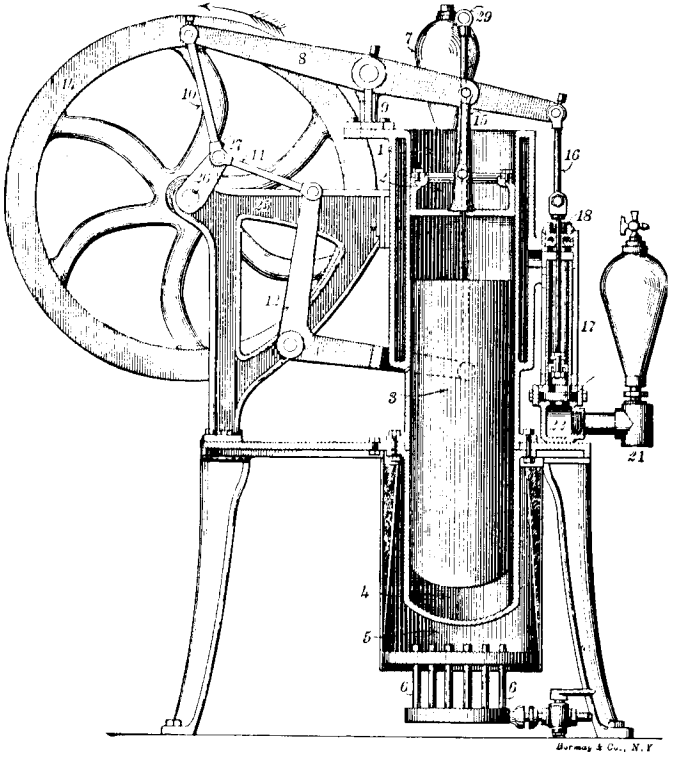

FIG. 72 —SECTION OF ERICSSON PUMPING ENGINE, WITH BUNSEN-BURNER GAS FURNACE.

1, Cylinder; 2, air piston; 3, transfer piston; 4, heater; 5, furnace; 6, gas burners; 7, air chamber; 8, main beam; 9, beam centre bearing; 10, connecting rod; 11, bell-crank link; 12, bell crank; 14, fly-wheel; 15, air piston links; 16, pump link; 17, pump chamber; 18, pump gland; 19, suction valve; 21, suction pipe; 22, pump bottom; 25, crank-shaft bracket; 26, crank; 27, crank pin; 29, transfer piston-rod, cross-head.

cylinder, as represented in Figs. 71 and 72. It operates without a special regenerator. The hot air from the heating chamber passes in a thin stratum along the outside of the transfer piston, and is cooled in its course toward the working piston by convection from the water-jacketed surface of the upper part of

the cylinder, the pumped water passing through the cylinder jacket for this purpose.

In the type of the Rider hot-air engine, operating through the same recurring cycle and at a constant initial volume with differential heat pressures, the extremes of heat and cold are established in different cylinders, the pistons being operated from a common shaft with cranks at right angles to meet the cyclic requirement, as shown in the cut (Fig. 73). The office of the regenerator is to intensify the extreme temperatures by absorbing much of the heat of the air as it passes from the heat cylinder to the cold cylinder, and to return the same heat to the air in its return from the cold to the hot cylinder. This operation gives a greater range to the temperature, and thereby increases the range of pressures.

Actual observation of the temperature at each end of the regenerator has shown a difference of 300° F., which indicates a considerable differential pressure, modified by the proportional part of the air volume in the two cylinders not acted upon by the regenerator. This may equal a mean differential temperature of 250° F. for the whole volume of the enclosed air. The respective volumes will then become as 1 to 1.27, and the pressures 0 to 4.23 pounds per square inch during a half revolution, with probably a mean pressure of 2½ pounds per square inch during a revolution of the fly-wheel. This will be equivalent to about 2,500 foot-pounds, minute, in a 5-inch engine, or nearly one-twelfth of a horse power.

The indicator card (Fig. 74), taken from a Rider two-cylinder hot-air engine by Professor Hutton, represents the cycle of pressures derived from the apparently erratic motion of cranks at right angles and operated by two pistons, both of which were under variable pressure from heat expansion in a constant initial volume of air.

The drop of the indicator line below the atmospheric line during a half-stroke indicates a leakage of air under the pressure of three half-strokes, or three-quarters of a revolution.

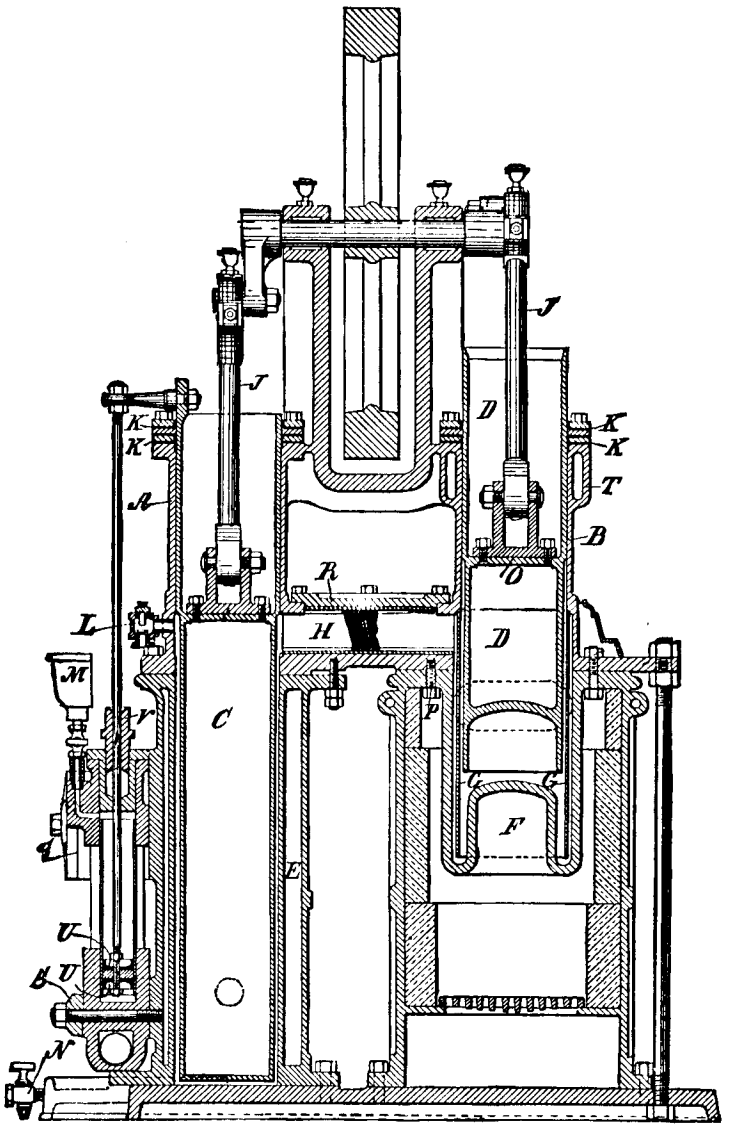

FIG. 73.—SECTION OF THE RIDER HOT-AIR PUMPING ENGINE.

A, Compression cylinder; *B*, power cylinder; *C*, compression piston; *D*, power piston *E*, cooler; *F*, heater; *G*, telescope; *H*, regenerator; *I I*, cranks; *J J*, connecting rods; *K K* piston packings; *L*, check valve, at back of compression cylinder; *M*, pump primer; *N*, blow off cock; *R*, regenerator bonnet; *S S*, pump-valve bonnet; *T*, water jacket, to protect packing from heat; *U U*, pump buckets; *V*, pump gland.

The hot-air engines of the Ericsson-Rider type do not operate on the constant-volume cycle; for the operation of the working pistons varies the relative volumes by the differential length of the cranks and consequent amount of the volume of the stroke, and also does not operate at constant pressure; hence, the heat volume of the air is variable. At constant pressure a motor piston cannot pass through a cyclical movement and do work. So that it becomes evident that in the investigation of the movement of the pistons of this class of engines, the volume and pressure are both variable, and that

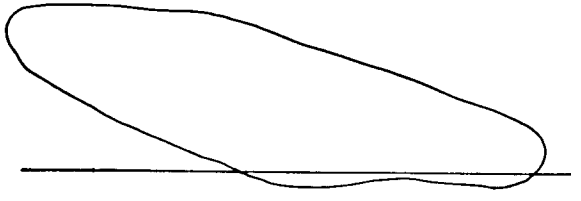

FIG. 74.—INDICATOR CARD.
Rider hot-air engine.

both volume and pressure are made variable by heat exchange and thus become the elements of motive power.

In the traverse of the two pistons in the Ericsson type the transfer piston is neutral in pressure, save the air friction; but in the Rider type the two pistons are of equal size, single acting, both working against the outer atmospheric pressure, and have the internal pressure equal on both pistons, save the air friction by transfer. Its power is derived from the differential stroke, the transfer piston having the longest stroke by about 16 per cent. The Ericsson hot-air pumping engine is made in four sizes, viz., 5-, 6- 8-, and 10-inch diameter of cylinder. The Rider hot-air pumping engine is made in five sizes, viz., 4-, 5-, 6-, 8-, and 10-inch diameter of cylinders.

Frank Richards, 1913

CHAPTER XVII

RE-HEATING COMPRESSED AIR

It is sufficiently well known that after the transmission of compressed air to the point where it is to be employed, it having lost its heat of compression on the way, or, indeed, having had that heat abstracted by aftercooling devices, a considerable saving in the cost of the available power may be effected, theoretically at least, by re-heating the air before it is put to do its work. We still, however, have little reliable practical data concerning this suggested economy, or actual knowledge of the precise conditions under which it becomes available.

The specific heat of air being low, comparatively little heat is required from an external source to raise the temperature of compressed air rapidly and, under constant pressure, to increase its volume correspondingly. While air at a low temperature has a comparatively small cooling effect upon water, or upon whatever may come in contact with it, the fact inversely applied may be taken advantage of in the use of air as a power transmitter.

It may easily be shown that where a certain volume of air has been compressed to any given pressure, and has, by aftercooling, by transmission or by storage, returned to approximately its normal temperature, if that air is then re-heated and thereby expanded, the additional volume of compressed air resulting is produced by a much lower expenditure of heat than the original volume of compressed air was produced for, and also by a much lower expenditure of heat than would be required to produce an equal working volume of steam. The actual figures in the case, all theoretical, are as follows:

Weight of 1 cu. ft. of steam at 75 lb. gage = 0.2089 lb.

Total units of heat in 1 lb. of steam at 75 lb. from water at 60° = 1151.

Total units of heat in 1 cu. ft. of steam at 75 lb. = 1151 × 0.2089 = 240.44.

To produce by compression through a steam-actuated air-

compressor 1 cu. ft. of compressed air at 75 lb and 60°, about 2 cu. ft. of steam of the same pressure are required, or the heat-units employed in producing 1 cu. ft. of compressed air will be about $240.44 \times 2 = 480.88$ heat units as the thermal cost of 1 cu. ft. of compressed air at the above temperature and pressure. The temperature and volume of the air as it leaves the compressor will be considerably higher than the figures here assumed, but as the air is invariably stored for a time, or is transmitted through pipes to a distance between its compression and its ultimate employment, it may be said to always return to its normal temperature before it is used, so that, whatever we may have at the compressor, the air as it is delivered to the motor, or whatever apparatus may be operated by it, will have cost, as above stated, 480.88 heat units for 1 cu. ft. at 75 lb. The difference in the thermal cost of any volume of compressed air thus produced by mechanical compression and the cost of any additional volume of air that may result from the subsequent re-heating of the air is very striking.

The weight of 1 cu. ft. of free air at $60° = 0.076$ lb.

Weight of 1 cu. ft. of compressed air at 75 lb. and $60° = 0.456$.

Units of heat required to double the volume of 1 lb. of air at $60° = 123.84$.

Units of heat required to double the volume of 1 cu. ft. of compressed air at 75 lb. and $60° = 123.84 \times 0.456 = 56.47$.

Cost of 1 cu. ft. of superheated compressed air at 75 lb. compared with the cost of 1 cu. ft. of compressed air as produced by ordinary compression:

$$480.88 : 56.47 :: 1 : 0.1174.$$

Here we see that the cost in heat-units of the volume of air produced by the re-heating is less than one-eighth of the cost of the same volume produced by compression. Upon this showing the matter is certainly worth looking into.

The operation of re-heating compressed air is correctly so termed. It is, in fact, a case of doing work over again, or of replacing in the air heat that has been lost by it in previous operations. It must be confessed that the presumption is all against our finding much profit in this direction. There are not many places in life where it pays to do our work a second time. There is, as we have seen, practically no air compression without heating the air by the operation, and there is no transmission of air after compression without its cooling to very near its original temperature. If the air could go immediately from the compressing cylinder into the motor cylinder, where it does its work, without losing any of its heat, it would have the same effective power as it would have after long-distance transmission and cooling and re-heating, and without the additional cost of that re-heating.

While we are saying in all good faith that there is little loss of power in the transmission of compressed air to considerable distances, and that the difference in the pressure of the air at the two ends of a long pipe necessary to overcome the friction and maintain the flow is but small, and that it is to a great extent compensated for by the increased volume at delivery, the fact still is that there is a great loss of power in the transmission of the air, if we reckon from the moment when compression ceases, on account of the inevitable cooling of the air. Still this loss is not properly chargeable to the transmission, for no matter how far the air may be transmitted the cooling is all accomplished before the air has travelled very far, if the pipes are of proper size. Supposing air to be transmitted 10 miles, it must be conveyed with considerable rapidity if it does not get down to normal temperature before the end of the first quarter of a mile.

As the volume of air under any constant pressure varies directly as the absolute temperature, it follows that to double the volume by heating the air its absolute temperature must be doubled. The air being at 60°, its absolute temperature will be $60 + 461 = 521$, and double this will be $521 \times 2 = 1042$, the absolute temperature required. This by the thermometer will be $1042 - 461 = 581°$. As this is the temperature that is required for the air when delivered into the motor, and actually beginning its work, it will be necessary, on account of the ease and rapidity with which it cools, to heat the air considerably higher than this theoretical temperature. It is one thing, and an easy one, to heat the air, while it is a very different and a very difficult thing to keep it hot. To avoid all loss of heat it would be necessary, not only to keep the pipe which conveyed the air constantly hot, but also the cylinder in which it was used, or it would be cooled before it began to do its work.

In one case within my experience, where compressed air was re-heated, and its absolute temperature was increased at the heater 38 per cent., and where, of course, its theoretical increase of volume was the same, the actual increase of power realized was only 12 per cent. In this case the air was transmitted after the re-heating about 20 ft., the pipe was not covered, and no precautions were taken to prevent loss of heat by radiation. The

[illegible] of air transmitted was sufficient to develop between 20 and 30 per cent. theoretical augmentation of temperature required to double the volume of compressed air at 60° being 581°, the actual temperature required at the heater under the most favorable conditions in order to have a double volume of air available at the motor will not be less than 800°, and this is a temperature that it is practically impossible to employ and maintain, lubrication troubles would defeat it if nothing else, and we may as well give up all thought of doubling the volume of compressed air by re-heating it and of realizing the promised economy of such an operation.

If instead of doubling the volume we only attempt to increase it by one-half, or 50 per cent., which it is practicable to do, the required theoretical temperature (absolute) will be 521+50 per cent. =782, and 782−461=321°, the sensible temperature required. Adding enough to this to allow for the intermediate cooling, the actual temperature required should probably be not less than 450°. The temperature of the air before the re-heating being assumed to be 60°, the increase of temperature will be 450°−60°=390°. As we saw above that it required 56.47 heat units to raise the temperature of 1 cu. ft. of compressed air at 75 lb. gage pressure from 60° to 581°, the actual increase of temperature being 581−60=521, it follows that to raise the temperature 390° will require:

$$521 : 390 : : 56.47 : 42.27.$$

Then if the first cubic foot of compressed air costs 480.88 heat units for its compression, and if the additional half of a cubic foot produced by re-heating costs 42.27 heat units, the total cost of 1 1/2 cu. ft. under the re-heating system will be 480.88+42.27 = 523.15, and the cost per cubic foot at this rate will be 523.1÷ 1 1/2 =348.76 heat units. The relative cost in heat units of 1 cu. ft. of compressed air produced by compression alone, and of a cubic foot resulting from compression and re-heating, will be:

$$480.88 : 348.76 : : 1 : 0.72.$$

From this it appears that the gain by re-heating compressed air sufficiently to increase its effective volume 50 per cent. will be 28 per cent. The more fair and correct way to state this, however, will be to reverse it:

$$0.72 : 1 : : 1 : 1.38.$$

We may say, then, that the total fuel applied with the re-heating system will yield 38 per cent. higher results than are to be realized without the reheating. This seems to be very near the maximum that can be attained in the way of economy by re-heating *dry* compressed air.

But, after all, it must be confessed that it is not always, nor indeed often, that the re-heating of compressed air is practicable or possible. Bearing in mind the facility and rapidity with which heated air in transmission loses its heat, it is idle to think of ever

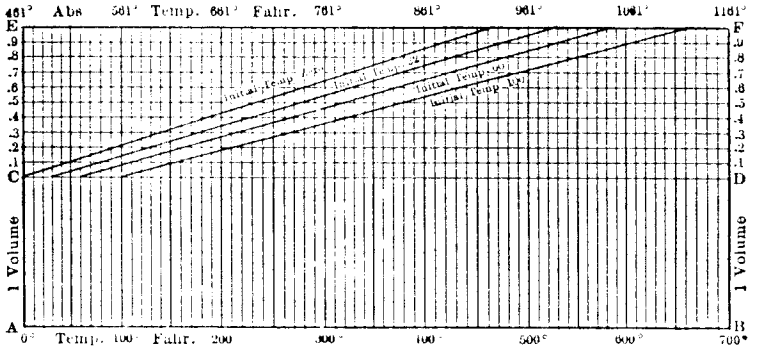

FIG. 52.—Increase of Volume by Reheating.

profitably re-heating compressed air except for continuously running motors, and then by heaters very close to the motors.

Wherever any motor or engine is to be run by compressed air without interruption, a heater for the air should certainly be employed; but there are now several hundred different and distinct uses of compressed air in not one of which it would be practicable or anything but a losing operation to try to heat the air.

Fig. 52 shows the increase of volume accompanying the heating or reheating of compressed air. The air is assumed to be heated from the several initial temperatures of 0°, 32°, 60°, and 100°, the pressure remaining constant during the operation represented. The relative volume at any temperature is indicated by the height of the vertical line corresponding with that temperature, the height from *AB* to *CD* representing one volume, and each horizontal line above that indicating, successively, an additional one-tenth of volume. When the line *EF* is reached, the original volume is doubled. Figures below the base-line *AB* indicate the sensible temperatures Fahrenheit, and the figures above the upper line indicate the corresponding absolute temperatures.

Fig. 53 shows the increase of pressure only caused by the heating of compressed air, the volume being constant. The air is assumed to be heated, as in Fig. 52, from the several initial temperatures of 0°, 32°, 60°, and 100°, and also from a number

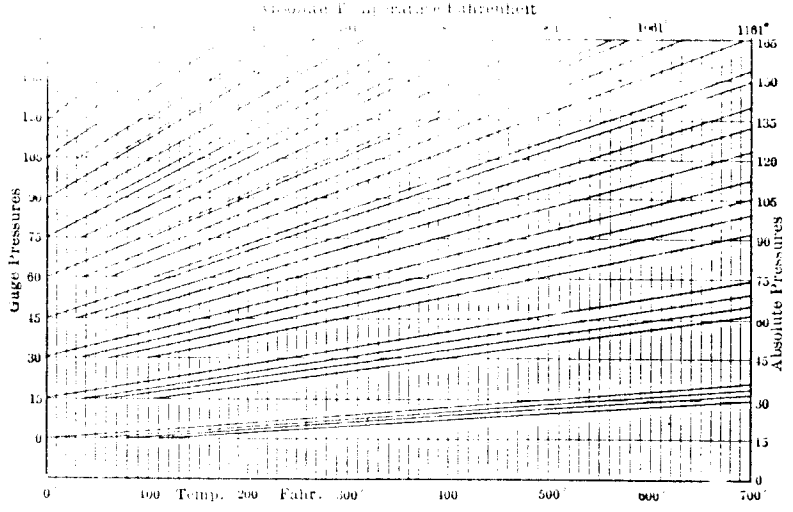

FIG. 53.—Increase of Pressure by Reheating.

of different initial pressures. The pressures are indicated by the several horizontal lines, the vertical distance between any two adjacent lines representing approximately 1 atmosphere. The figures at the left of the diagram indicate the gage pressures, and the figures at the right the absolute pressures. The temperatures are indicated as in the previous diagram.

It looks as if the most unsatisfactory phase of the re-heating problem to-day is in the re-heater itself. There are re-heaters so-called of various makes on the market, but where is *the* re-heater? It is not enough to provide a receptacle where fuel may be burned and then sufficient heating surface over which, or heated tubes through which, the air to be heated may be passed. This might do when the air was flowing steadily all day long, as for instance in driving a mine pump, but where else of all the places in which compressed air is used?

Supposing the heater to be of the right capacity and the fire adjusted to give the required air temperature when flowing at a certain rate, then if the air flowed faster it would not be heated enough, and if it flowed slower it would be overheated. If the flow stopped entirely and then after a time started again the air might be so hot as to burn the packing or make other trouble in that way. In either case there would be waste of fuel and the economy sought would be only partially realized, if at all.

Then with this arrangement who is to know what the actual temperature of the air is, except by watching a recording thermometer and then only to realize that the heat is not right and is not steady at any figure? The proper working temperature for the air should be decided upon before hand, and then the ideal reheater should maintain this temperature constant, within narrow limits, just as an intercooler or an aftercooler maintains an approximately constant temperature when working in the opposite direction. These coolers deliver the air at a temperature approximating that of the circulating water whose temperature varies only imperceptibly; so it would appear that if water were employed as the circulating medium, and if this water were kept at a constant temperature then the temperature of the air heated by it could be depended upon to be nearly constant.

The temperature of the water in a steam boiler rises or falls with the steam-pressure, and if the steam-pressure remains constant the water temperature coincides with that of the steam —or rather *vice versa*. By carrying a proper steam-pressure the right water temperature for re-heating compressed air may be obtained and maintained; and this steam pressure need not be maintained with minute precision, as the temperature changes are not rapid. Thus with 100-lb. gage pressure the steam and water temperature would be a trifle under 340° while at 200 lb. the temperature would be 50° higher, and anywhere within this range the air would be heated to above 300°.

A more constant steam-pressure and a steadier temperature may be maintained than here suggested, as is often done in house-heating arrangements, by means of automatic dampers and other fire-controlling devices, without blowing off or wasting the steam, or the hot water behind it, and all of the heat of the fire actually used would be that which went to the heating of the air.

The better arrangement would be to have a large tubulated steam space and a small water space and to heat the air by the steam rather than by the water. The smaller body of water would enable it to be heated and steam generated quicker, with better command of the temperature fluctuations.

It may be possible that there are other combinations equally effective for re-heating compressed air, but that here suggested has the advantage of having been successfully employed in at least one notable instance. At the admirable compressed-air installation at the mines of the Anaconda Copper Mining Company at Butte, Montana, where compressed air has supplanted steam for the driving of the big hoisting engines as spoken of in the preceding chapter, the air is re-heated by steam, and the additional power gained by the re-heating is stated to be obtained at a cost of one-third of a pound of coal per horse-power hour.

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THE REHEATING OF COMPRESSED AIR

BY

CHARLES RUSS RICHARDS

DIRECTOR OF THE ENGINEERING EXPERIMENT STATION AND DEAN OF THE COLLEGE OF ENGINEERING

AND

JOHN N. VEDDER

RESEARCH ASSISTANT IN MECHANICAL ENGINEERING

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THE REHEATING OF COMPRESSED AIR

I. INTRODUCTION

1. *Preliminary Statement.*—For many years compressed air has been employed as a medium for the transmission of power. The simplicity and relatively high efficiency of air compressors, the economy with which compressed air may be transmitted for comparatively long distances through pipe lines, and the variety of air motors and tools now available have led to the extensive use of compressed air in mining, quarrying, and tunneling, in various shop processes, and in the development of power.

Many devices operated by compressed air are run intermittently; and with such apparatus economy of operation is generally sacrificed for simplicity and minimum initial cost of equipment. Few air-driven tools use the air expansively, wherefore their thermal efficiency is small. When motors are operated continuously for considerable periods of time, however, economy of operation becomes a consideration, and it is essential that the air be expanded in the motor. Since the variations of volume and pressure of air during compression and expansion may be represented by the equation $pv^n = a$ constant, air is heated during compression and cooled during expansion. In the compressed air plant the heat of compression is quickly dissipated by radiation from the air receivers and pipe lines, so that when the air reaches the motor its temperature is little if any higher than that of the atmosphere. Consequently the temperature of the air after expansion is so low that it causes serious operating difficulties, resulting both from the freezing of the moisture in the air and the consequent accumulation of ice in the exhaust pipe, and from the

ineffectiveness of cylinder lubrication and resultant loss of mechanical efficiency. These difficulties may be overcome by heating the air before its delivery to the motor to an initial temperature such that the temperature of the exhaust air will be above the freezing point of water. This heating process is commonly designated as "reheating," and the term will therefore be employed in the discussion which follows.

It should be noted that when compressed air is used as a medium for the transmission of power the reheating process affords a means of increasing the total output of power per pound of air through the expenditure of a small additional amount of heat energy, and that no other system of power transmission offers similar advantages.

2. *Purposes of Investigation and Significant Results.*—This investigation of the reheating of compressed air was undertaken to determine the ideal thermodynamic efficiencies resulting from the heat expended in the reheating process, the efficiency of external and internal combustion reheaters, the performance of an engine using air expansively under a wide variety of operating conditions, and the performance of the same engine operating with steam alone, and with mixtures of air and steam when steam is injected into the air pipe as a means of reheating the air.

The results of the investigation of reheaters show that in small external combustion reheaters maximum efficiencies of from 16.7 to 61.5 per cent may be secured, depending upon the type of reheater employed; and that in the internal combustion reheater of the type tested the efficiency varied from 69.4 per cent when 326 pounds of air per hour was heated, to 83.0 per cent when 1240 pounds of air per hour was heated.

The tests of the engine using air as the working medium show that under the most favorable conditions maintained the expenditure of coke per indicated horse power per hour gained through reheating was 0.98 pounds, and, per brake horse power per hour gained, 0.90 pounds. Under the least favorable conditions of operation with cold air the mechanical efficiency of the engine fell to 59.8 per cent, while under the best conditions of operation with reheated air the mechanical efficiency was 93.7 per cent.

In the tests of the engine using a mixture of air and steam it was found that the work done per pound of mixture was considerably in excess of that attainable by the separate use of the same weight of each ingredient in the mixture. The interest in air-steam mixtures developed during this investigation has led to further studies of the subject, the results of which will be presented in a later bulletin.

3. *Acknowledgments.*—The experimental work of this investigation was carried on in the Power Laboratory of the Department of Mechanical Engineering by JOHN N. VEDDER,* Research Assistant in Mechanical Engineering. During a part of the time, he was assisted by WILLIAM E. MOSHER,† Research Graduate Assistant in Mechanical Engineering. Because of uncertainty regarding the accuracy of the method originally employed in calibrating the Venturi tube used as an air meter during the tests, the tube was recently recalibrated and all air weights recalculated by CRANDALL Z. ROSECRANS, Research Assistant in Mechanical Engineering. This recalibration was performed with the air weighing plant described on page 81 of Engineering Experiment Station Bulletin No. 120, entitled "Investigation of Warm Air Furnaces and Heating Systems."

The authors are indebted to the Ingersoll-Rand Company, the Sullivan Machinery Company, and the Kreutzburg Manufacturing Company for the loan of reheaters used during the investigation.

* Now Associate Professor of Thermodynamics at Union College, Schenectady, New York.

† Mr. Mosher died in Washington, D. C., on October 12, 1918. At that time he was Assistant Superintending Engineer of the U. S. Army Transport Service, in charge of marine refrigeration.

II. THERMODYNAMICS OF REHEATED AIR: IDEAL CYCLES

4. *Case 1. No Expansion.* Compressed air may be used in motors in which there is no expansion of the air, in those in which there is partial expansion, and in those in which there is complete expansion. The ideal efficiency resulting from the use of reheated air—that is, the ratio of the extra theoretical work secured through the use of reheated air, expressed in British thermal units (B.t.u.), to the heat added to reheat the air, in B.t.u.—is here developed for these three cases.

In Fig. 1 let $p_e = p_a = p_b =$ the initial pressure in pounds per square foot of air admitted to the cylinder of a motor whose clearance

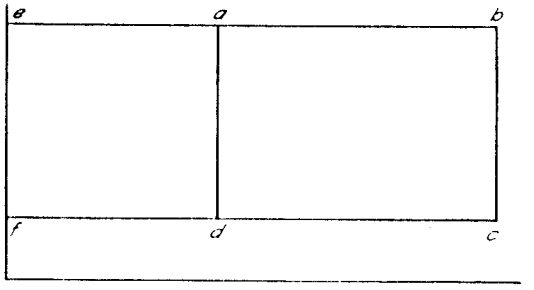

FIG. 1. WORK DIAGRAM FOR AIR. NO EXPANSION

volume is zero; and $p_f = p_d = p_c$ = the pressure of the atmosphere. Assume that v_a = the volume in cubic feet of one pound of air whose absolute temperature is T_a. The theoretical work in foot-pounds per pound of air at T_a is represented by the area ***e a d f***. If the air is heated at constant pressure to a temperature T_b its volume will become v_b and the theoretical work per pound of air is represented by the area *e b c f*. The difference between these areas, *a b c d*, represents the additional work per pound of air resulting from the heat added as "reheat."

The area $a\,b\,c\,d = (p_a - p_d)(v_b - v_a)$ ft. lbs.

$= A\,(p_a - p_d)(v_b - v_a)$ B.t.u.

when $A = \frac{1}{778}$ = the reciprocal of the mechanical equivalent of heat.

The heat supplied $= c_p\,(T_b - T_a)$ B.t.u.

$= c_p \frac{T_a}{v_a}(v_b - v_a)$ B.t.u.,

since $\frac{T_a}{v_a} = \frac{T_b}{v_b}$

and c_p = mean specific heat of air at constant pressure.

The efficiency ratio then becomes

$$E = \frac{A\,(p_a - p_d)\,(v_b - v_a)}{c_p \frac{T_a}{v_a}(v_b - v_a)} = \frac{A\,v_a\,(p_a - p_d)}{c_p\,T_a}$$

$\frac{p_a\,v_a}{T_a} = B$, the air constant.

If c_v = the mean specific heat of air at constant volume

$$B = \frac{c_p - c_v}{A}$$

and $$\frac{v_a}{T_a} = \frac{B}{p_a} = \frac{c_p - c_v}{A\,p_a}$$

Hence $$E = \frac{k-1}{k}\left(1 - \frac{p_d}{p_a}\right) = 0.2884\left(1 - \frac{p_d}{p_a}\right)$$

since $k = \frac{c_p}{c_v} = 1.405$

5. *Case II. Complete Expansion.*—In Fig. 2 let $p_e = p_a = p_b =$ the initial pressure; $p_f = p_d = p_c =$ the atmospheric pressure in pounds per square foot; and $v_a =$ the volume in cubic feet of one pound of air at a temperature T_a and pressure p_a. If in a cylinder without clearance the air be expanded to the atmospheric pressure p_d, the work in foot pounds for the cycle is represented by the area ***e a d f***. If the air is reheated at constant pressure from T_a to T_b, its volume is increased from v_a to v_b; and the work of the cycle for complete expansion is represented by the area *e b c f*. The difference between the areas, *a b c d*, represents the additional work

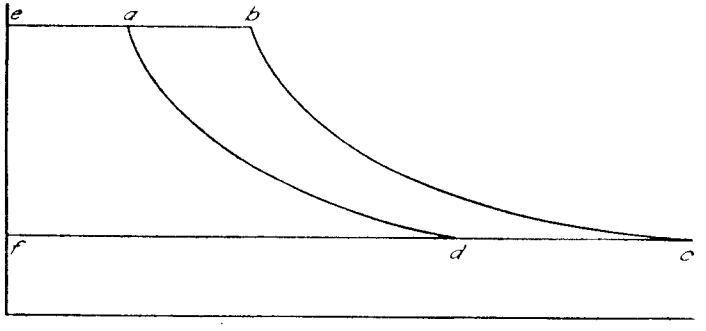

Fig. 2. Work Diagram for Air. Complete Expansion

per pound of air secured through the heat added. The efficiency of the reheating process is the ratio of the heat converted into work to the heat added. With complete expansion of the air, it is obvious that the efficiency is the maximum attainable.

Since $p_a\,v_a^n = p_d\,v_d^n$

and $p_b\,v_b^n = p_c\,v_c^n$

the work area $e\,a\,d\,f = \frac{n}{n-1}\,p_a\,v_a\left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right]$ ft. lbs.,

and the work area $e\,b\,c\,f = \frac{n}{n-1}\,p_b\,v_b\left[1 - \left(\frac{p_c}{p_c}\right)^{\frac{n-1}{n}}\right]$ ft. lbs.,

$= \frac{n}{n-1}\,p_a\,v_b\left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right]$ ft. lbs.

The work gained $a\,b\,c\,d = \frac{n}{n-1}\,p_a\,(v_b - v_a)\left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right]$ ft. lbs.,

$= \frac{n}{n-1}\,B\left(T_b - T_a\right)\left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right]$ ft. lbs.

The "reheat" supplied $= \frac{1}{A}\,c_p\left(T_b - T_a\right)$ ft. lbs.

The efficiency then becomes

$$E = \frac{n}{n-1}\left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right]\frac{A\ B}{c_p}$$

$$= \left(\frac{n}{n-1}\right)\left(\frac{k-1}{k}\right)\left[1 - \left(\frac{p_d}{p_a}\right)^{\frac{n-1}{n}}\right]$$

$$= \left(\frac{n}{n-1}\right)\left(\frac{k-1}{k}\right)\left[1 - \left(\frac{v_a}{v_d}\right)^{n-1}\right]$$

$$= \left(\frac{n}{n-1}\right)\left(\frac{k-1}{k}\right)\left[1 - \frac{T_d}{T_a}\right]$$

If the air is expanded adiabatically n becomes equal to k and the efficiency is reduced to

$$E = 1 - \left(\frac{p_d}{p_a}\right)^{\frac{k-1}{k}} = 1 - \left(\frac{v_a}{v_d}\right)^{k-1} = 1 - \frac{T_d}{T_a}$$

6. *Case III. Partial Expansion.*—The least favorable case for reheated air used expansively is shown by the diagram, Fig. 3. As

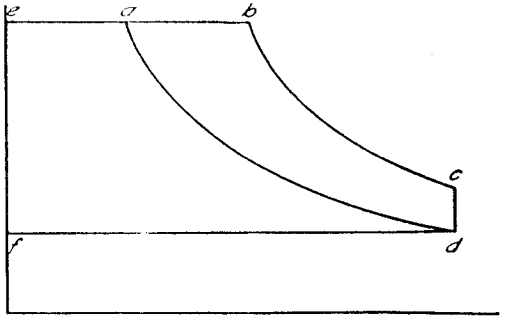

Fig. 3. Work Diagram for Air. Partial Expansion

in Case II, the area $e\ a\ d\ f$ represents the work in foot pounds per pound of air whose initial volume is v_a at a pressure p_a and temperature T_a. Assume that the temperature is increased to T_b at constant pressure, and that the volume v_c at the end of expansion is the same as the volume v_d attained after expansion of the unreheated air to atmospheric pressure. It is evident that the work area is the smallest possible when expansion is employed if the final volume is not reduced below that attained when cold air is used. If the expansion of the reheated air is carried farther than shown, that is, if v_c is greater than v_d, the thermal efficiency will be somewhere between that of Case II and Case III.

In Fig. 3 the work resulting from the reheat supplied is represented by the area $a\ b\ c\ d$, which is the difference between areas $e\ b\ c\ d\ f$ and $e\ a\ d\ f$. Since

$$\text{area } e\ b\ c\ d\ f = \frac{n}{n-1}(p_a v_b - p_c v_c) + v_c (p_c - p_d)$$

$$\text{and area } e\ a\ d\ f = \frac{n}{n-1}(p_a v_a - p_d v_c)$$

$$\text{area } a\ b\ c\ d = \frac{n}{n-1}\left[(p_a v_b - p_c v_c) - (p_a v_a - p_d v_c)\right] + v_c (p_c - p_d)$$

$$= \frac{n}{n-1} p_a (v_b - v_a) - \frac{1}{n-1} v_c (p_c - p_d) \text{ ft. lbs.}$$

Substituting the appropriate values of $pv = BT$ in the preceding equation and multiplying by A,

$$\text{the area } a\ b\ c\ d = \frac{AB}{n-1}\left[n(T_b - T_a) - (T_c - T_d)\right] \text{ B.t.u.}$$

The reheat supplied $= c_p (T_b - T_a)$; therefore the efficiency is

$$E = \frac{k-1}{k(n-1)}\left(n - \frac{T_c - T_d}{T_b - T_a}\right)$$

Since $$T_d = T_a\left(\frac{v_a}{v_d}\right)^{n-1}$$

$$T_b = T_a\left(\frac{v_b}{v_a}\right)$$

and $$T_c = T_b\left(\frac{v_b}{v_d}\right)^{n-1} = T_a \frac{v_b}{v_a}\left(\frac{v_b}{v_d}\right)^{n-1}$$

$$E = \frac{n(k-1)}{k(n-1)}\left[1 - \frac{1}{n}\left(\frac{v_a}{v_d}\right)^{n-1}\frac{\left(\frac{v_b}{v_a}\right)^n - 1}{\frac{v_b}{v_a} - 1}\right]$$

If adiabatic expansion is assumed, $n = k$ and the expression for efficiency reduces to

$$E = 1 - \frac{1}{k}\left(\frac{v_a}{v_d}\right)^{k-1}\frac{\left(\frac{v_b}{v_a}\right)^k - 1}{\frac{v_b}{v_a} - 1}$$

In this case the efficiency decreases as the volume v_b and temperature T_b increase.

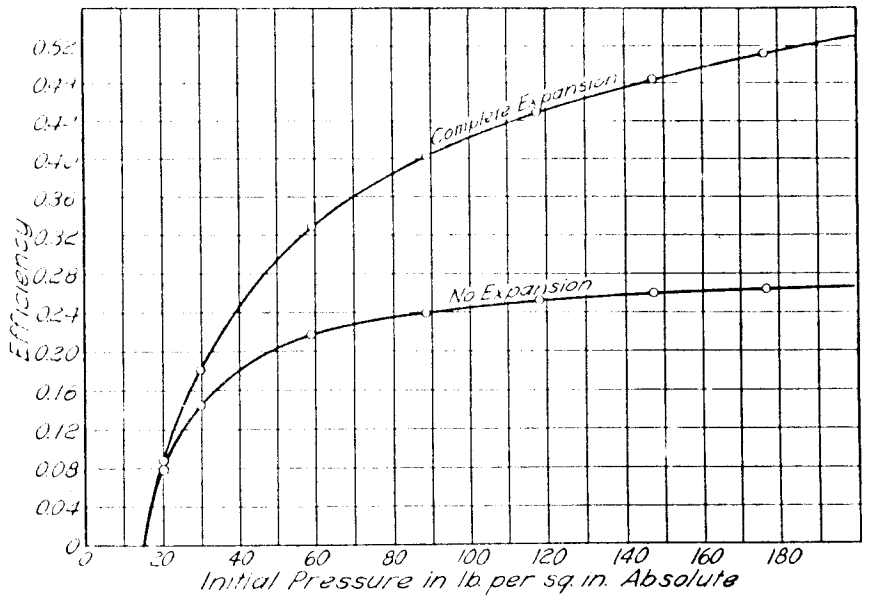

FIG. 4. THEORETICAL EFFICIENCIES OF AIR CYCLES

Reference to the efficiencies theoretically attainable in Cases I and II shows that they are independent of the reheat added and that they are functions of $\frac{p_d}{p_a}$ only. Fig. 4 shows the theoretical efficiencies for cycles having no expansion of the air and for those having complete adiabatic expansion.

III. TESTS OF REHEATERS

7. *Method of Testing External Combustion Reheaters.*—Most of the tests of external combustion reheaters had a duration of eight hours, all conditions during that period being maintained as nearly constant as possible. Since the reheaters tested and the weight of fuel consumed were small, the percentage of error in fuel weights due to starting and stopping the tests would have been increased with a reduction in time. With eight-hour tests, however, it was felt that this error would be very small.

Before starting a test a preliminary run of about one hour was made to insure the maintenance of standard conditions from the beginning of the test. When the compressor, piping system, and reheater had reached thermal equilibrium, the fire was quickly drawn and rebuilt with weighed kindling. This process required about three minutes, during which the air was closed off from the reheater and the output of the compressor escaped through the relief valve on the air receivers. As soon as the fire was properly started the air was turned on and the test begun.

The fuel used, a coke of good quality, was fired in approximately equal weights, and at equal intervals of time during each test. The weights in different tests varied with the weight of air heated; and, in general, the interval of time was 15 to 20 minutes, except in the case of the first two or three firings where the interval was shorter. During the preliminary run the depth of the fire necessary for the assigned "load" was determined; then, at the beginning of the test, the fire was brought to this depth in the shortest possible time by using maximum draft, until the flue gas temperature was 100 to 150 degrees above the normal for the test. This building up period lasted 15 to 20 minutes and the second and third charges were fired as soon as the previous charge showed red on top.

The constancy of the temperature of the reheated air leaving the reheater was the regulative factor controlling the tests. To maintain a constant temperature, a controllable furnace draft was found indispensable. The draft in the stack, to which the reheater was connected, was induced and controlled by a jet of compressed air from a one-eighth inch pipe. A valve controlling the jet of air made it possible to obtain any draft from 0.05 to 0.40 inches of water. A thermometer inserted through a stuffing box directly in the stream of heated air made it possible to observe the temperature and to maintain it, with a variable draft, to within about 10 degrees Fahrenheit of the mean temperature.

A valve in the air pipe about ten feet from its entrance to the reheater was employed to maintain in the reheater a constant pressure of 100 to 110 pounds per square inch, gage, during most of the tests. The pressure in the receiver was 10 to 15 pounds higher. The flow of air was controlled by a second valve in the pipe leaving the reheater. The proper openings of these valves were determined by trial during the preliminary run, but slight readjustments were frequently necessary.

To assist in regulating the draft the temperature of the flue gases was determined by a Hoskins pyrometer. This temperature was kept as low as possible to permit the maintenance of the desired temperature of the reheated air. Samples of flue gas were taken in aspirator bottles over four-hour periods and analyzed during or immediately after the test by means of an Orsat apparatus filled with freshly prepared solutions.

After leaving the reheater the air passed through about 12 feet of uncovered pipe to a gage-box fitted with a standard orifice. A pressure of from 2 to 5 inches of water was maintained in the box and the discharge was computed by using the coefficients recommended

by R. J. Durley.* As the relatively high temperatures might affect the coefficients of discharge, the Venturi tube, later used in metering the air for the engine tests, was connected in series with the reheater and gage-box, and several tests were run with air at different temperatures. As the ratio between the flow indicated by the meter and the gage-box followed the same law with and without the heater in operation, the use of Durley's coefficients for the measurement of air in the test was justified.

This same arrangement of apparatus was employed in calibrating the Venturi tube used in the tests. This tube is of a standard form manufactured by the Builders' Iron Foundry of Providence, Rhode Island. It is flanged for standard 2½-inch pipe connections and has a throat diameter of 0.75 inches. The standard differential mercury manometer supplied by the manufacturers of the tube was used in all tests. In certain air-steam mixture tests, made at a later date than those described in this bulletin, the question of the accuracy of the calibration of the Venturi tube was raised and various attempts were made to recalibrate the apparatus by methods other than the one originally employed. No one of these attempts was successful until use was made of the air weighing plant* used to calibrate instruments employed in the investigation of warm air furnaces. A careful recalibration of the Venturi tube indicated that, with Durley's coefficients, the original calibration against standard orifices gave weights of air which were 3 per cent low. Accordingly all computations involving weights of air originally made for this bulletin have been corrected to agree with this new calibration. This merely involved the multiplication of the weights of air as originally calculated by 1.03. The determination by means of the Venturi tube of the weight of air flowing involves much laborious computation. In order to reduce the labor and at the same time secure greater accuracy, large scale graphical charts, prepared for use with the Venturi tube, were employed in making the calculations. It seems unnecessary to present this material here.†

* Durley, R. J. "On the Measurement of Air Flowing into the Atmosphere through Circular Orifices in Thin Plates and Under Small Differences of Pressure." Trans. A. S. M. E., v. 27, p. 193, 1906.

* For a description of the plant see "Investigation of Warm Air Furnaces and Heating Systems," Univ. of Ill. Eng. Exp. Sta., Bul. No. 120, 1921.

† The general theory of gaseous flow through Venturi tubes is presented in a paper, "The Flow of Fluids in a Venturi Tube," by Edgar P. Coleman. Trans. A. S. M. E., v. 28 p. 183.

During the tests the following observations were recorded every 10 minutes:

(1) Temperature of air entering reheater
(2) Pressure of air entering reheater
(3) Temperature air leaving reheater
(4) Flue gas temperature
(5) Pressure in gage-box
(6) Temperature in gage box
(7) Room temperature
(8) Draft

Other observations and records were:

(9) Barometric pressure (each hour)
(10) Diameter of orifice
(11) Position of damper
(12) Weight of kindling
(13) Time of each firing
(14) Net weight of fuel on scales after each firing
(15) Weight of fuel fired each firing
(16) Weight of residual fuel
(17) Weight of ash and refuse
(18) Flue gas analysis

Samples of each load of fuel placed on the scales were taken and sealed for analysis. Fig. 5 shows the general arrangement of testing apparatus, the figures referring to the numbered observations just listed. The fire was kept up to the standard condition until the end of the test. About two minutes before this time, the grates were thoroughly shaken and the ash and refuse collected, weighed, and preserved. Then at the end of the test the grates were dumped, and the fire quenched with the minimum quantity of water necessary. The residual fuel was then collected, weighed, and preserved.

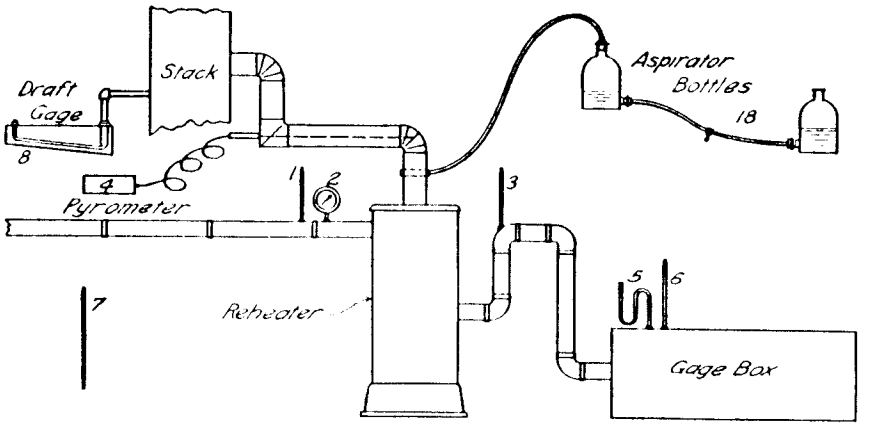

FIG. 5. GENERAL ARRANGEMENT OF APPARATUS FOR TESTING EXTERNAL COMBUSTION REHEATER

8. *Definitions and Notation Employed.*—In rating compressed air equipment it is customary to express capacity in cubic feet of free air per minute. The term "free air" is loosely used, and without specific definition is meaningless. As employed in this bulletin the free air unit is one cubic foot of air at a temperature of 62 degrees Fahrenheit and a pressure of 14.7 pounds per square inch absolute. The volume of a pound of air at 62 degrees Fahrenheit and 14.7 pounds per square inch absolute pressure is 13.147 cubic feet. The weight of a cubic foot of free air is 0.07608 pounds. One pound of air per hour is equivalent to 0.21906 cubic feet of free air per minute.

The specific heat of dry air at constant temperature, c_p, does not greatly vary within the temperature range employed in this investigation, but when moisture is present, the change of specific heat for small temperature changes is marked. According to Swann, the specific heat at constant pressure of dry air at 20 degrees centigrade (68 degrees Fahrenheit) is 0.24301. In computations made in this investigation the value of c_p for dry air was taken as 0.243, and it was assumed that the air before reheating was saturated with water vapor. By the theory of mixture of gases and vapors it is possible to calculate the value of c_p for such mixtures. Values of c_p thus calculated are given in the various tables presented.

The efficiency of a reheater is the ratio of the heat absorbed by the compressed air to the calorific value of the fuel charged against the reheater. The numerator of this fraction is the product of the weight of air heated in pounds, the specific heat of the air at constant pressure, and the rise in temperature in degrees Fahrenheit; the denominator is the calorific value of the total fuel fired minus the calorific value of the residual fuel either with or without the fuel in the ash—depending upon whether the efficiency of the grate is excluded or included. In internal combustion reheaters using liquid fuel the denominator is the calorific value of the liquid fuel consumed.

For convenience of reference the designations of all tests are explained as follows:

Reheater tests are designated by two numbers and a letter. The first number denotes the approximate quantity of free air in cubic feet per minute; the second, the approximate temperature of the air after reheating; and the letter is added only for the sake of distinction; e.g., 150–400–a.

Engine tests using air are designated by four numbers: first, a serial number; second, the gage pressure at the throttle; third, the temperature at the throttle; and fourth, the approximate cut-off in per cent of the engine stroke; e.g., 50–100–200–25.

Air-steam engine tests are denoted by the letters A–S followed by the gage pressure at the throttle and the approximate percentage of steam in the mixture; e.g., A–S–100–10.

Tests with steam alone are designated by the letter S, followed by the gage pressure at the throttle; e.g., S–80.

9. *Tests of Type I Reheater.*—This reheater consists of a cast iron fire-pot in the form of the frustrum of a cone, with base downward, and surrounded by a shell with an annular space through which the air circulates. A detachable grate at the bottom of the fire-pot made it possible to draw the fire in a few seconds. Since the air heating surface completely surrounds the fire-pot, the temperature of the exterior of the reheater is not greater than that of the reheated air; hence the loss by radiation is small. Type I reheater has an effective heating surface of 2.9 square feet, and, when reheating air to 400 degrees Fahrenheit, a capacity of about 200 cubic feet of free air per minute.

Coke was used as a fuel in all external combustion reheater tests. The samples of coke, residual fuel, ash, and refuse collected during the several tests of the Type I reheater were analyzed by the Department of Chemistry of the University of Illinois with the following results:

		B.t.u.
(1)	Calorific value per pound of air-dry coke as fired, by the oxygen calorimeter	12 992
(2)	Proximate analysis of air-dry coke:	
		Per Cent
	Fixed carbon	88.76
	Volatile matter	1.65
	Moisture	0.55
	Ash	9.04
		100.00
	Moisture on drying	0.00
	Sulphur determined separately	0.25
(3)	Ultimate analysis of air-dry coke:	
	Carbon	87.21
	Hydrogen	0.74
	Oxygen	1.34
	Nitrogen	0.87
	Sulphur	0.25
	Ash	9.04
	Moisture at 105 deg. C.	0.55
		100.00
(4)	Proximate analysis of residual fuel:	
	Carbon	86.30
	Earthy matter	13.40
	Moisture	0.30
		100.00

(5) Proximate analysis of ash and refuse:

Carbon	30.92
Earthy matter	68.93
Moisture	0.15
	100.00

It was assumed that each pound of wood used as kindling had a calorific value of 0.4 pound of coke, and that each pound of oil used in kindling the fire had a calorific value of 1.47 pound of coke. The calorific value of carbon is taken as 14 540 B.t.u. per pound. From these assumptions and the data from the several analyses the various items presented in Table I were computed. In test 100–300–b the weight of ash and refuse was not recorded. Since they were obtained by computing the ash and refuse from the average grate efficiency, items 18 and 20 for this test are enclosed in brackets to indicate that their accuracy is open to some question.

Table 2 presents various data and the economic results of the tests of Type I reheater. The several numbered columns of the table present average recorded data or computed results. The methods of computing the various items are either indicated or obvious without further explanation. The efficiencies shown under items 14 and 15 are of chief interest. The efficiencies, exclusive of grate, vary from 48.0 to 61.5 per cent. The best efficiencies were obtained when from 100 to 150 cubic feet of free air per minute were reheated.

10. *Tests of Type II Reheater.*—In the Type II reheater the compressed air flowed through a series of cast iron coils set in a casing so that the coils were 12 to 15 inches above the normal level of the fire. No baffling was provided to force the heated gas over the coils, and the area of the furnace and casing exposed to the outer air was large; consequently the loss of heat by radiation was great. The heating surface of the coils was 18.6 square feet. The tests of this reheater were made in exactly the same manner as has been described for the Type I reheater.

TABLE 1

RESULTS OF TESTS OF TYPE I REHEATER

Designation	Gage-box Data						Gross Fuel per Test			
	1	2	3	4	5	6	7	8	9	10
	Date	Length of Test (hrs.)	Diameter of Orifice (in.)	Pressure in Gage-box (in. Water)	Temperature in Gage-box (deg. F.)	Barometer (in. Mercury)	Gage-box Coefficient (Durley's Curves)	Coke (lb. per Test)	Wood (lb. per Test)	Oil (lb. per Test)
50–400–a	6–10–12	8.00	1.5	3.56	241	29.5	0.603	24.0	3.00	0.30
100–400–a	6–11–12	8.00	2.0	4.86	297	29.4	0.601	42.5	3.00	0.25
150–400–a	6–12–12	8.00	2.5	4.59	316	29.2	0.598	52.0	3.00	0.25
200–400–a	6–14–12	8.00	3.5	2.04	308	28.8	0.597	70.0	4.00	0.30
275–200–b	5–30–12	6.33	3.5	3.88	203	29.3	0.595	39.5	3.00	0.66
275–200–c	6– 1–12	8.50	3.5	4.02	223	29.1	0.595	50.0	3.00	0.60
100–250–b	6– 3–12	8.00	2.5	2.15	220	29.2	0.599	23.0	3.00	0.40
200–300–b	6–13–12	4.00	3.5	2.16	287	29.1	0.597	33.0	3.00	0.30
100–300–b	5–27–12	5.00	2.5	2.08	225	28.8	0.599	20.0	1.25	0.10
100–300–c	5–29–12	8.00	2.5	2.06	243	29.1	0.599	37.5	3.00	0.30
100–400–b	6– 5–12	8.00	2.5	2.10	291	29.3	0.599	44.0	3.00	0.30

Designation	Fuel Equivalents						Net Fuel			
	11	12	13	14	15	16	17	18	19	20
	Ash and Refuse (lb. per Test)	Residual Fuel (lb. per Test)	Coke Equivalent of Wood and Oil (lb. per Test)	Total Equivalent Coke (lb. per Test) (Item 8 + Item 13)	Coke Equivalent of Ash and Refuse (lb. per Test)	Coke Equivalent of Residual Fuel (lb. per Test)	Net Equivalent Coke Charged against Reheater (lb. per Test) (Item 14 − Item 16)	Net Equivalent Coke Consumed (lb. per Test) (Item 17 − Item 15)	Net Equivalent Coke Charged against Reheater (lb. per hr.) (Item 17 ÷ Item 2)	Net Equivalent Coke Consumed (lb. per Test) (Item 18 ÷ Item 2)
50–400–a	1.50	4.50	1.64	25.64	0.52	4.36	21.28	20.76	2.66	2.60
100–400–a	2.50	11.00	1.56	44.06	0.87	10.70	33.36	32.49	4.17	4.06
150–400–a	4.50	8.25	1.56	53.56	1.56	7.94	45.62	44.06	5.71	5.51
200–400–a	7.75	12.00	2.03	72.03	2.71	11.66	60.37	57.66	7.55	7.21
275–200–b	4.42	13.11	2.16	41.66	1.54	12.72	28.94	27.40	4.56	4.33
275–200–c	4.50	9.75	2.07	52.07	1.56	9.49	42.58	41.02	5.02	4.82
100–250–b	1.75	3.00	1.78	24.78	0.61	2.92	21.86	21.25	2.74	2.66
200–300–b		9.50	1.64	34.64	9.70	9.23	25.41	24.71	6.35	6.18
100–300–b		7.88	0.65	20.65	...	7.66	13.00	[12.55]	2.60	[2.51]
100–300–c		11.23	1.64	39.14	0.94	10.93	28.21	27.27	3.53	3.41
100–400–b		11.00	1.64	45.64	6.87	10.70	34.94	34.07	4.37	4.26

TABLE 2

RESULTS OF TESTS OF TYPE I REHEATER

Designation	1 Length of Test (hr.)	2 Pressure of Air Entering Reheater (lb. per sq. in. Gage)	3 Temperature Air Entering Reheater (deg. F.)	4 Temperature Air Leaving Reheater (deg. F.)	5 Rise of Temperature	6 Air Reheated (lb. per hr.)	7 Air Reheated in cu. ft. of Free Air per min. (62° F., 14.7 lb. Abs.)	8 Specific Heat at Const. Press. Air Saturated Entering Reheater	9 B. t. u. Transmitted to Compressed Air (per hr.)	10 B. t. u. Transmitted per hr. per sq. ft. of Heating Surface	11 Total B. t. u. Supplied per hr.	12 Grate Loss per hr. in B. t. u.
50–400–a	8.00	109.7	98.1	402.8	304.7	218	47.7	0.244	16170	5590	34559	844
100–400–a	8.90	109.9	111.5	409.2	297.7	432	95.0	0.244	31440	10860	54250	1406
150–400–a	8.00	109.6	121.0	397.2	276.2	644	141.0	0.245	43750	15110	73665	2530
200–400–a	8.00	117.4	132.7	385.9	252.3	838	183.5	0.245	51810	17890	98112	4303
275–200–b	6.33	99.4	128.1	225.0	96.9	1253	275.0	0.246	29810	10290	58675	2388
275–200–c	8.50	100.1	139.5	247.3	107.8	1232	271.0	0.246	33130	11420	67530	2380
100–250–b	8.00	108.0	111.2	278.7	167.5	471	103.0	0.244	19250	6640	35545	984
200–300–b	4.00	108.5	126.3	343.8	217.2	880	192.5	0.245	46750	16140	82627	2250
100–300–b	5.00	95.7	108.8	287.0	178.2	459	100.0	0.244	19960	6880	33820	[1210]
100–300–c	8.00	101.4	101.6	324.4	222.8	454	99.4	0.244	24620	8520	45896	1525
100–400–b	8.00	106.9	110.1	401.5	291.4	444	97.5	0.244	31570	10870	56805	1406

Designation	13 B. t. u. Liberated per hr. by Combustion	14 Efficiency Including Grate (Per cent)	15 Efficiency Excluding Grate (Per cent)	16 Net Equivalent Coke Charged Against Reheater (lb. per hr.)	17 Pounds Coke Required to Raise 1000 lb. Air 100° F. (Item 16) + $\left(\frac{\text{It.5}}{100} \times \frac{\text{It.6}}{1000}\right)$	18 Draft At Base of Stack in in. of Water	19 Room Temperature (deg. F.)	20 Barometer (in. Mercury)	21 Temperature of Flue Gases (deg. F.)	22 Flue Gas Analysis		
										Carbon Dioxide (CO_2)	Oxygen (O_2)	Carbon Monoxide (CO)
50–400–a	33715	46.8	48.0	2.66	4.00	0.09	80	29.5	568	8.9	10.8	0.0
100–400–a	52844	57.9	59.4	4.17	3.24	0.15	84	29.4	611	9.1	10.1	0.0
150–400–a	71135	59.4	61.5	5.71	3.20	0.24	87	29.2	581	10.8	8.1	0.0
200–400–a	93749	52.8	55.2	7.55	3.57	0.35	84	28.8	601	10.5	8.4	0.0
275–200–b	56287	50.6	53.0	4.56	3.75	0.04	75	29.3	572	12.8	5.7	0.0
275–200–c	65150	49.1	50.9	5.02	3.77	0.03	85	29.1	561	13.2	5.0	0.0
100–250–b	34561	54.4	55.6	2.74	3.47	0.08	83	29.2	606	6.5	12.7	0.5
200–300–b	80377	56.6	58.1	6.35	3.32	0.30	80	29.1	682	9.5	9.5	0.0
100–300–b	[32610]	59.0	[61.1]	2.60	3.18	0.05	86	28.8	475	....	....	....
100–300–c	44371	52.4	55.6	3.53	3.50	0.05	73	29.1	607	11.3	7.7	0.0
100–400–b	55399	55.6	57.0	4.37	3.38	0.14	81	29.3	629	12.7	6.8	0.5

Analyses of the fuel, residual fuel, ash, and refuse gave the following results:

		B.t.u.
(1)	Calorific value per pound of air-dry coke as fired, by the oxygen calorimeter	12 718
(2)	Proximate analysis of air-dry coke:	Per Cent
	Fixed Carbon	87.25
	Volatile matter	2.24
	Ash	0.50
	Moisture	10.01
		100.00
	Moisture on drying	0.00
	Sulphur separately determined	0.66
(3)	Ultimate analysis of coke:	
	Carbon	83.95
	Hydrogen	1.20
	Oxygen	2.68
	Nitrogen	1.00
	Sulphur	0.66
	Ash	10.01
	Moisture at 105 deg. C.	0.50
		100.00
(4)	Proximate analysis of residual fuel:	
	Carbon	58.64
	Earthy matter	41.22
	Moisture	0.14
		100.00
(5)	Proximate analysis of ash and refuse:	
	Carbon	18.62
	Earthy matter	81.38
	Moisture	0.00
		100.00

Tables 3 and 4 present all the essential data and results of the tests of this reheater. The efficiencies obtained were low, due to excessive loss of heat by radiation and in the flue gases. With small loads the fire was necessarily thin and the excess air large; with larger loads the loss from excess air was reduced, although the flue gas temperature was abnormally high, and the loss of heat in the

TABLE 3

RESULTS OF TESTS OF TYPE II REHEATER

Designation	Gage-box Data							Gross Fuel per Test		
	1	2	3	4	5	6	7	8	9	10
	Date	Length of Test (hrs.)	Diameter of Orifice (in.)	Pressure in Gage-box (in. Water)	Temperature in Gage-box (deg. F.)	Barometer (in. Mercury)	Gage-box Coefficient (Durley's Curves)	Coke (lb. per Test)	Wood (lb. per Test)	Oil (lb. per Test)
50–400–d	6–22–12	8.00	1.5	3.58	242	29.5	0.602	65.0	10.00	0.25
100–400–d	6–24–12	8.00	2.0	4.80	301	29.3	0.600	100.0	6.00	0.75
150–400–d	10–23–12	8.00	3.0	2.19	311	29.4	0.598	103.5	8.00	1.50
200–400–d	10–22–12	8.00	3.0	3.84	308	29.2	0.597	133.5	9.50	1.50
250–400–d	10–24–12	8.00	3.5	3.27	325	29.4	0.597	145.5	8.75	1.50
300–400–d	10–25–12	8.00	3.5	4.75	341	29.3	0.595	174.5	9.50	1.50
350–400–d	10–26–12	8.00	4.0	3.96	348	29.3	0.595	189.5	9.50	1.50

TABLE 3 (CONTINUED)

RESULTS OF TESTS OF TYPE II REHEATER

Designation	Fuel Equivalents						Net Fuel			
	11	12	13	14	15	16	17	18	19	20
	Ash and Refuse (lb. per Test)	Residual Fuel (lb. per Test)	Coke Equivalent of Wood and Oil (lb. per Test)	Total Equivalent Coke (lb. per Test) (Item 8 + Item 13)	Coke Equivalent of Ash and Refuse (lb. per Test)	Coke Equivalent of Residual Fuel (lb. per Test)	Net Equivalent Coke Charged against Reheater (lb. per Test) (Item 14 − Item 16)	Net Equivalent Coke Consumed (lb. per Test) (Item 17 − Item 15)	Net Equivalent Coke Charged against Reheater (lb. per hr.) (Item 17 ÷ Item 2)	Net Equivalent Coke Consumed (lb. per hr.) (Item 18 ÷ Item 2)
50–400–d	4.75	10.00	4.36	69.36	1.01	6.71	62.65	61.64	7.83	7.71
100–400–d	6.40	14.70	3.49	103.49	1.35	9.88	93.61	92.26	11.70	11.53
150–400–d	6.50	8.50	5.63	109.12	1.38	5.70	103.42	102.04	12.93	12.76
200–400–d	8.00	11.00	5.69	139.19	1.70	7.39	131.80	130.10	16.48	16.26
250–400–d	6.50	14.00	5.68	151.18	1.38	9.41	141.77	140.39	17.72	17.55
300–400–d	8.00	21.50	5.98	180.48	1.70	14.42	166.06	164.36	20.76	20.55
350–400–d	10.00	15.00	5.98	195.48	2.12	10.03	185.45	183.33	23.18	22.92

TABLE 4

RESULTS OF TESTS OF TYPE II REHEATER

Designation	1	2	3	4	5	6	7	8	9	10	11	12
	Length of Test (hr.)	Pressure of Air Entering Reheater (lb. per sq. in. Gage)	Temperature Air Entering Reheater (deg. F.)	Temperature Air Leaving Reheater (deg. F.)	Rise of Temperature	Air Reheated (lb. per hr.)	Air Reheated in cu. ft. of Free Air per min. (62° F., 14.7 lb. Abs.)	Specific Heat at Const. Press. Air Saturated Entering Reheater	B. t. u. Transmitted to Compressed Air (per hr.)	B. t. u. Transmitted per hr. per sq. ft. of Heating Surface	Total B. t. u. Supplied per hr.	Grate Loss per hr. in B. t. u.
50–400–d	8.00	82.3	98.3	390.6	292.3	232	51	0.242	16397	882	99602	1606
100–400–d	8.00	82.8	105.4	406.0	300.6	428	94	0.243	31286	1682	148860	2166
150–400–d	8.00	104.4	109.1	405.6	296.5	661	145	0.243	47640	2561	164425	2200
200–400–d	8.00	102.3	118.8	380.2	261.4	868	190	0.245	55604	2989	209556	2716
250–400–d	8.00	106.6	122.2	392.4	270.2	1078	236	0.245	71353	3837	225425	2200
300–400–d	8.00	106.7	136.1	399.5	263.4	1279	280	0.246	82844	4454	263994	2766
350–400–d	8.00	106.5	146.4	398.0	251.6	1513	333	0.247	94060	5058	294775	3387

Designation	13	14	15	16	17	18	19	20	21	22		
										Flue Gas Analysis		
	B. t. u. Liberated per hr. by Combustion	Efficiency Including Grate (Per cent)	Efficiency Excluding Grate (Per cent)	Net Equivalent Coke Charged Against Reheater (lb. per hr.)	Pounds Coke Required to Raise 1000 lb. Air 100° F. (Item 16 ÷ [It.5/100 × It.6/1000])	Draft At Base of Stack (in. Water)	Room Temperature (deg. F.)	Barometer (in. Mercury)	Temperature of Flue Gases (deg. F.)	Carbon Dioxide (CO_2)	Oxygen (O_2)	Carbon Monoxide (CO)
50–400–d	97996	16.5	16.7	7.83	11.50	0.14	89	29.5	415	4.6	16.3	0
100–400–d	146694	21.0	21.3	11.70	9.10	0.14	100	29.3	495	6.0	14.0	0
150–400–d	162225	29.0	29.4	12.93	6.60	0.18	67	29.4	574	5.6	14.2	0
200–400–d	206850	26.5	26.9	16.48	7.27	0.35	73	29.2	599	4.5	15.5	0
250–400–d	223225	31.7	32.0	17.72	6.08	0.21	70	29.4	[illegible]	8.3	11.6	0
300–400–d	261288	31.4	31.7	20.76	6.15	0.20	76	29.3	[illegible]	9.7	10.6	0
350–400–d	291388	31.9	32.3	23.18	6.09	0.24	75	29.3	[illegible]	11.4	8.8	0

flue gas large. While the efficiencies might have been somewhat improved by substituting a long flaming coal for coke, the general design of the reheater is not conducive to fuel economy.

11. *Tests of Type III Reheater.* This reheater is of the internal combustion type using gasoline as fuel. Its capacity is about 250 cubic feet of free air per minute, heated to a temperature of 400 degrees Fahrenheit. The sectional view of the reheater, Fig. 6, shows

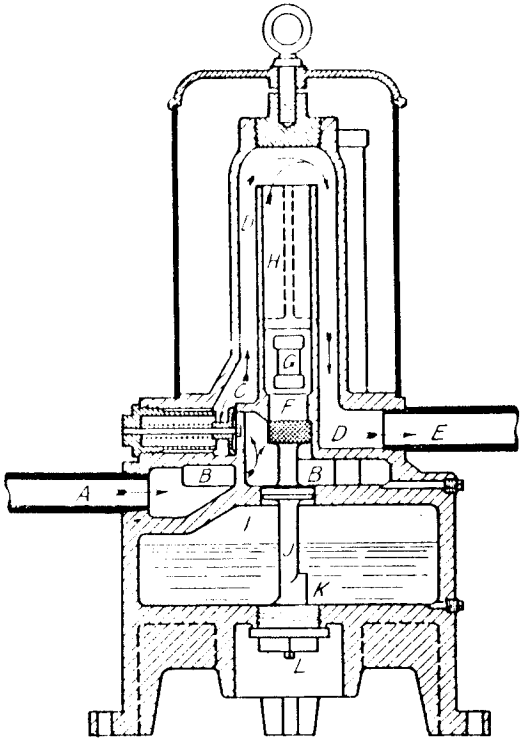

FIG. 6. SECTIONAL VIEW OF INTERNAL COMBUSTION REHEATER

the method of operation. Cold air enters the reheater at *A* and the reheated air leaves at *E*. The incoming air enters the space *B* and its pressure is exerted on the surface of the gasoline, *I*, in the base of the reheater. *C* is an adjustable spring loaded valve set to cause a slight reduction of pressure in the air, which flows through the valve into the space *D* and out at *E*. A tube, *J*, having an opening, *K*, from the gasoline reservoir leads to a needle valve and atomizer at *F*, adjustable at *L*; above the atomizer a combustion tube opens directly into the space *D*. Since the air pressure on the surface of the gasoline is slightly above that in space *D* and tube *H*, gasoline is forced through the atomizing tube upon the electrical ignition coil, *G*. The gasoline mixture burns in *H* and the products of combustion pass out with the reheated air.

In actual operation it was found difficult to secure and maintain the proper adjustment of the differential valve, *C*, and the atomizer. Considerable difficulty with the electrical ignition device was also experienced. Despite these difficulties, however, this reheater was used throughout the various engine tests described in a later section, and in general it proved satisfactory and convenient. It is decidedly easier to maintain constancy of reheat with this reheater than with the external combustion type. The slightest change in the position of the needle valve controlling the flow of gasoline affected at once the temperature of the out-going air.

The tests of this reheater were relatively simple. The Venturi tube already described was employed to measure the air. Its arrangement is shown in Fig. 7, and that of the reheater in Fig. 8.

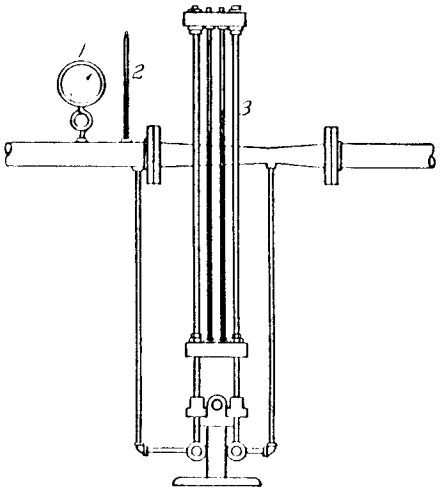

FIG. 7. APPARATUS FOR TESTING INTERNAL COMBUSTION REHEATER; GENERAL ARRANGEMENT OF VENTURI TUBE

The numbers shown in these two figures refer to the various observations necessary, as follows:

(1) Upstream pressure on tube (every 10 minutes)
(2) Upstream temperature (every 10 minutes)
(3) Manometer (every 2 minutes)
(4) Temperature of air entering reheater (every 10 minutes)
(5) Pressure of air entering reheater (every 10 minutes)
(6) Pressure of air leaving reheater (every 10 minutes)
(7) Temperature of air leaving reheater (every 5 minutes)
(8) Gasoline gage at beginning and end of test
(9) Room temperature (once for each test)
(10) Barometer (once for each test)

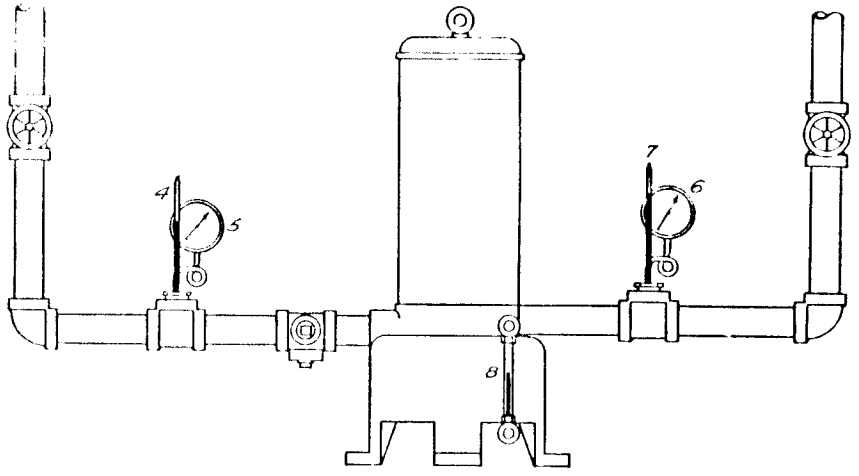

FIG. 8. APPARATUS FOR TESTING INTERNAL COMBUSTION REHEATER; GENERAL ARRANGEMENT OF REHEATER

With three exceptions the duration of each test was one hour; two tests were continued for 1½ hours, and one—an overload test designated as 370–400–f— was stopped after 30 minutes because of the excessive load on the compressor. The gasoline used in the tests of this reheater had a specific gravity of 0.732 and a calorific value of 20 309 B.t.u. per pound.

Fig. 9 shows the drop of pressure through the reheater at different rates of air flow. During test 370–400–f, the compressor was unable to deliver the air at the normal pressure of 100 pounds gage;

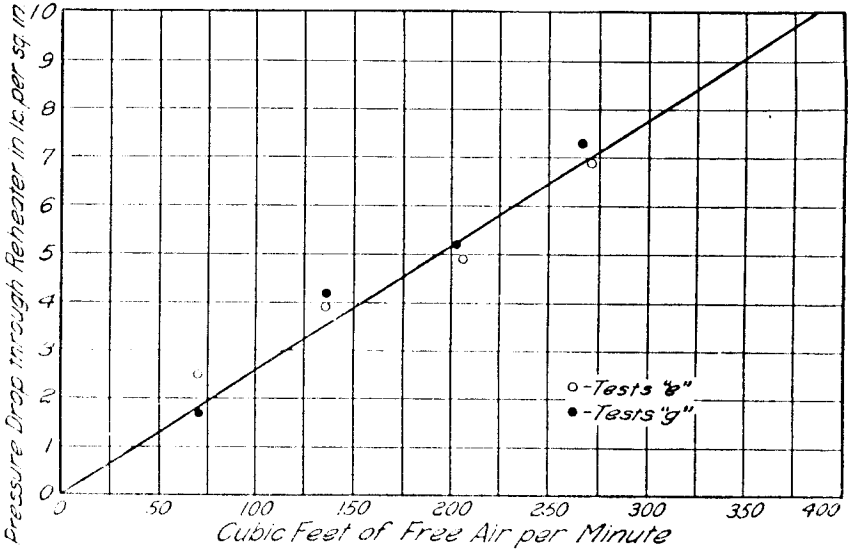

FIG. 9. DROP OF PRESSURE IN REHEATER

in fact the average pressure was only 48 pounds gage. This probably had an effect on the loss of pressure with an air flow of 370 cubic feet of free air per minute.

Table 5 presents the various observed quantities and the computed results. Item 17, the efficiency of the reheater, shows that due to more perfect vaporization and combustion of the gasoline the efficiency is better for the larger loads and higher temperatures of the reheated air. During some of the engine tests described in a later section, the temperature of the reheated air was maintained at 200 degrees Fahrenheit, and in some instances with comparatively small rates of flow. Under these conditions the combustion was so poor that there was a strong odor of unburned gasoline in the engine exhaust. Since the loss of heat by radiation could not have been large during any of the tests, it is evident that under the most favorable conditions maintained the loss from incomplete combustion was excessive.

IV. TESTS OF ENGINE USING AIR AS WORKING FLUID

12. *Purposes of Tests.*—A series of tests, during which air at various initial temperatures was used expansively as the working fluid in a small Corliss engine, was made to determine first, what proportion of the heat applied in the reheating process is utilized in the development of indicated work within the cylinder, and as effective or "brake" work delivered by the engine; and second, the efficiencies of the actual engine using reheated air in comparison with those theoretically obtainable. To secure the desired results it was necessary to establish the efficiencies of the engine under various conditions of operation without reheat, and to compare these results with those obtained under similar conditions of operation with the air heated to the several temperatures, specified by the range of the investigation. Since the injection of steam in the air pipe has occasionally been employed as a means of reheating the air in order to overcome operating difficulties with the motor, the experimental investigation was extended to determine the effectiveness of air-steam mixtures in comparison with steam and unreheated air used separately.

13. *Description of Experimental Plant.*—Air for the tests was supplied by an Ingersoll-Sergeant, two-stage, cross compound compressor, having an approximate capacity of 375 cubic feet of free air per minute compressed to 100 pounds per square inch gage. From the compressor the air was delivered to a receiver having a capacity of 65 cubic feet. From there it passed through the Venturi tube previously described, to a second receiver—capacity, 17 cubic feet—employed to reduce the pulsations of pressure in the Venturi

TABLE 5

RESULTS OF TESTS OF TYPE III REHEATER

Designation	1	2	3	4	5	6	7	8	9	
	Length of Test (hr.)	Pressure of Air Entering Reheater (lb. per sq. in. Gage)	Pressure of Air Leaving Reheater (lb. per sq. in. Gage)	Pressure Drop Through Reheater	Temperature Air Entering Reheater (deg. F.)	Temperature Air Leaving Reheater (deg. F.)	Rise of Temperature	Specific Heat at Const. Press. Air Saturated Entering Reheater	Venturi Meter Data	
									Temperature Upstream (deg. F.)	Pressure Upstream (lb. per sq. in. Gage)
70–400–e	1.5	99.3	96.8	2.5	99.0	402.0	303.0	0.244	112	99.4
140–400–e	1.0	99.5	95.6	3.9	112.3	403.0	290.7	0.245	132	99.7
200–400–e	1.0	99.7	94.8	4.9	125.3	400.0	274.7	0.247	154	100.0
260–400–e	1.5	100.8	93.9	6.9	140.0	405.0	265.0	0.250	168	101.8
370–400–f	0.5	48.0	34.5	13.5	156.0	404.0	248.0	0.264	180	52.8
70–300–g	1.0	99.1	97.4	1.7	110.0	300.2	190.2	0.244	122	100.4
140–300–g	1.0	100.5	96.3	4.2	113.1	302.1	189.0	0.245	133	100.5
200–300–g	1.0	100.0	94.8	5.2	128.4	302.2	173.8	0.247	148	100.0
260–300–g	1.0	99.3	92.0	7.3	137.4	308.2	170.8	0.250	165	99.4

Designation	9	9	10	11	12	13	14	15	16	17	18
	Venturi Meter Data		Air Reheated (lb. per hr.)	Air Reheated (cu. ft. of Free Air per min.) (62° F., 14.7 lb. Abs.)	B. t. u. Absorbed by Compressed Air (per hr.)	Fuel Used (lb. per hr.)	Lb. of Gasoline Required to Raise 1000 lb. of Air 100° F.*	Coke Equivalent of Item 14 (calorific Value 13 000 B. t. u. per lb.)	B. t. u. in Fuel Used (per hr.)	Efficiency (Per cent)	Room Temp. (deg. F.)
	Manometer (in. Mercury)	Barometer (in. Mercury)									
70–400–e	0.35	29.2	328	69.7	24215	1.57	1.58	2.46	31884	75.9	92
140–400–e	1.32	29.2	624	137.0	44455	2.70	1.49	2.33	54834	81.1	97
200–400–e	3.21	29.2	940	206.0	63807	3.72	1.44	2.25	75549	84.5	99
260–400–e	5.91	29.3	1240	271.9	82159	4.73	1.44	2.25	96062	85.5	98
370–400–f	26.72	29.3	1727	378.0	113020	6.80	1.58	2.46	138100	81.9	98
70–300–g	0.35	29.1	326	71.4	15105	1.04	1.68	2.62	21121	71.5	83
140–300–g	1.30	29.1	621	136.2	28760	1.77	1.50	2.35	35947	80.0	83
200–300–g	3.05	29.1	920	201.8	39532	2.55	1.59	2.49	51788	76.3	83
260–300–g	5.81	29.1	1214	266.0	51853	3.47	1.67	2.61	70492	73.5	83

*(Item 13) + $\left(\frac{\text{Item 7}}{100}\right) \times \left(\frac{\text{Item 10}}{1000}\right)$

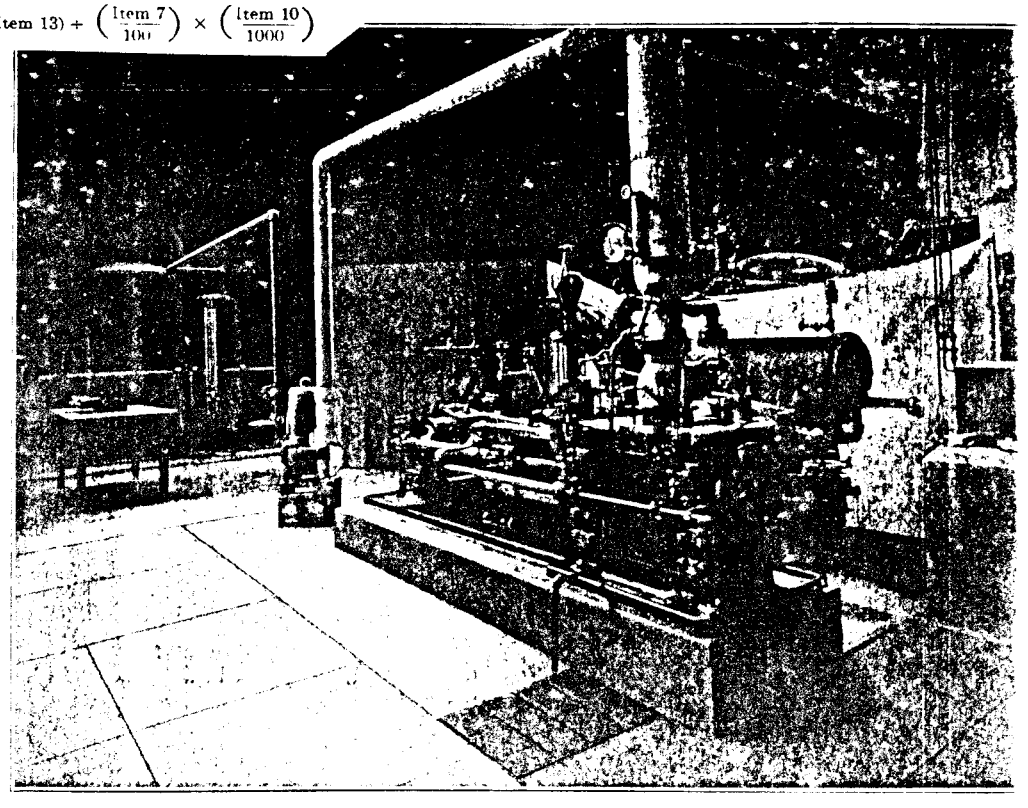

FIG. 10. GENERAL ARRANGEMENT OF EXPERIMENTAL PLANT

tube, and from thence through the internal combustion reheater, described in Section 11, to the engine. During the tests using air only the exhaust air from the engine was allowed to escape into the laboratory through a short exhaust pipe; during the tests of air-steam mixtures the exhaust pipe was connected to a Dean surface condenser having a tube surface of 100 square feet. Above the engine throttle valve the air line was provided with an effective mixing chamber and water separator for use when steam was injected into the air.

The engine used in this investigation was of the simple Corliss type built by the Murray Iron Works of Burlington, Iowa. Its principal dimensions are as follows:

	Inches
Diameter of Cylinder	8.00
Length of Stroke	18.05
Diameter of Piston Rod	1.87
Diameter of Steam Pipe	2.50
Diameter of Exhaust Pipe	3.00

Cylinder clearance in per cent of the total piston displacement is:

	Per Cent
For the Head End of the Cylinder	5.48
For the Crank End of the Cylinder	5.63

A specially designed indicator reducing motion of great accuracy was permanently attached to the engine, which was also equipped with a positive revolution counter and a Prony brake of excellent design and unusual steadiness. A pair of Tabor outside spring indicators were used in all tests.

Wherever necessary, pressures were determined by carefully calibrated standard test gages graduated in one pound increments. Temperatures were determined by standardized thermometers, those at the throttle and in the steam chest being secured by thermometers immersed through stuffing boxes directly in the stream of air. The temperature of the exhaust was secured by a pentane thermometer graduated in degrees centigrade, and placed in a mercury well in the exhaust pipe as close to the cylinder as possible. These temperatures are not an accurate measure of the actual temperatures in the cylinder during the exhaust stroke, since heat was rapidly absorbed from the surrounding atmosphere by the exhaust air in the pipe. During the earlier tests a hand pump was employed for lubricating the cylinder; in the later, a forced feed lubricator was attached to the engine. Fig. 10 shows the general arrangement of the plant.

14. *Method of Conducting Test.*—To insure a condition of thermal equilibrium in the piping and engine, the plant was operated for a period of one and one-half to two hours at the load to be maintained during the test. The cylinder of the engine was slow in reaching uniform temperature conditions, particularly when the point of cut-off was less than one-quarter of the stroke. When the difference between the temperatures at the throttle valve and in the steam chest became nearly constant it was possible to begin the actual test. As soon as thermal equilibrium was obtained, the predetermined conditions of pressure, air temperature, cut-off, etc., to be maintained during the test were established and the test was started. The order in which tests were undertaken was so chosen as to require the least possible change in conditions between tests, for any change involved the lapse of considerable time between tests to re-establish thermal equilibrium. The tests using air were continued for thirty minutes, during which the following measurements were secured:

(1) Upstream temperature of air (every 10 minutes)
(2) Temperature entering reheater (every 10 minutes
(3) Temperature leaving reheater (every 10 minutes)
(4) Temperature in steam chest (every 10 minutes)
(5) Temperature of exhaust (every 10 minutes)
(6) Temperature of room (every 10 minutes)
(7) Upstream pressure (every 10 minutes)
(8) Manometer reading of Venturi meter (every 2 minutes)
(9) Indicator diagrams (every 5 minutes)
(10) Height of barometer (once for each test)
(11) Height of gasoline in reservoir of reheater at beginning and end of test

The pressure and temperature of the air at the throttle and the load on the brake were adjusted at the beginning of the test and maintained constant during the test.

When the reheater was not in use the air reached the engine from the compressor at about 100 degrees Fahrenheit; therefore, this was established as the basic temperature of the unreheated air. It was originally intended to cool the air in the receiver to room temperature by a spray of water, but this plan was abandoned when it was found that with air at 100 degrees Fahrenheit, great trouble was experienced through the formation of ice in the exhaust ports and pipe.

15. *Classification of Air Tests.*—For convenience the meter, indicator, brake horse power, and reheater data for all the tests in which air alone was used are given in Table 6. Of the 63 tests reported those numbered 1 to 16, inclusive, are tests in which the expansion of the air was complete to atmospheric pressure; those numbered 17 to 22, inclusive, are the first tests with variable cut-off; those numbered

TABLE 6

OBSERVED VALUES FOR ENGINE TESTS WITH REHEATED AIR

		Venturi Meter				Indicator Data				b. h. p. Data		Reheater		
	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Designation	Date	Temperature of Air Upstream (deg. F.)	Pressure of Air Upstream (lb. per sq. in. Gage)	Barometer (in. Mercury)	Venturi Meter Manometer (in. Mercury)	m. e. p. Head End of Cylinder	m. e. p. Crank End of Cylinder	i. h. p. Head End of Cylinder	i. h. p. Crank End of Cylinder	Revolutions per min.	Weight on Scales	Temperature Entering Reheater (deg. F.)	Temperature Leaving Reheater (deg. F.)	Rise of Temperature (deg. F.)
1-100-400-15	2-19-13	132	103	29.0	0.83	28.0	29.8	6.80	6.86	106.1	116	104	452	348
2-100-300-15	2-19-13	135	103	29.0	0.91	27.7	28.9	6.73	6.65	106.1	111	103	328	225
3-100-200-15	2-19-13	115	103	29.0	1.04	27.6	29.7	6.71	6.83	106.1	106	97	215	118
4-100-100-15	2-20-13	130	103	29.2	1.54	28.6	31.2	6.94	7.17	105.9	101	....	....	....
5-80-400-20	2-24-13	101	82	29.4	0.87	26.5	27.4	6.40	6.26	105.4	106	80	446	366
6-80-300-20	2-24-13	90	82	29.4	1.00	26.3	26.9	6.35	6.16	105.5	101	74	338	264
7-80-200-20	2-21-13	126	83	29.0	1.07	25.9	26.5	6.27	6.08	105.8	86	92	227	135
8-80-100-20	2-20-13	116	83	29.2	1.80	27.7	28.3	6.68	6.47	105.3	90	....	....	....
9-60-400-25	2-24-13	94	62	29.4	0.93	23.0	23.8	5.50	5.39	104.4	91	82	453	371
10-60-300-25	2-21-13	105	62	28.7	1.11	23.0	23.1	5.51	5.24	104.6	86	88	325	237
11-60-200-25	2-21-13	123	63	29.0	1.34	23.2	23.4	5.57	5.33	104.9	81	93	232	139
12-60-100-25	2-20-13	137	62	29.2	1.93	22.7	23.7	5.45	5.39	104.8	71	....	....	....
13-40-400-35	2-24-13	93	43	29.4	1.08	18.5	19.6	4.40	4.41	103.7	72	81	472	391
14-40-300-35	2-21-13	102	42	28.9	1.22	17.7	18.4	4.21	4.14	103.9	64	85	328	243
15-40-200-35	2-21-13	107	43	29.0	1.49	18.7	18.8	4.46	4.25	104.2	56	89	246	157
16-40-100-35	2-20-13	130	41	29.2	1.92	17.7	18.3	4.22	4.14	104.2	48	....	....	....
17-100-400-15	3-11-13	138	103	29.3	0.90	30.3	30.3	7.33	6.94	105.6	123	102	443	341
18-100-400-17.5	2-25-13	118	103	29.3	1.00	33.0	33.5	7.99	7.68	105.7	136	98	447	349
19-100-400-20	3-15-13	150	105	28.8	1.30	37.9	37.4	9.13	8.54	105.2	153	114	437	323
20-100-400-20	2-25-13	119	103	29.3	1.26	39.1	38.6	9.42	8.80	105.2	156	98	443	345
21-100-400-25	2-25-13	119	104	29.4	1.65	45.1	44.7	10.82	10.16	104.7	181	98	438	340
22-100-400-25	3-15-13	153	104	28.8	1.74	43.8	45.8	10.45	10.36	104.2	181	112	435	323
23-100-400-27.5	2-25-13	120	104	29.4	1.93	48.5	48.7	11.59	11.04	104.4	200	93	445	352
24-100-400-30	3-11-13	124	105	29.3	2.15	49.3	51.4	11.74	11.60	104.0	209	100	439	339
25-100-400-30	3-15-13	151	104	28.8	2.21	50.9	50.9	12.12	11.49	104.0	206	107	435	328
26-100-400-35	3-11-13	137	106	29.3	2.78	57.1	56.1	13.55	12.61	103.6	236	103	439	336
27-100-300-25	3-19-13	135	104	29.1	2.00	43.2	45.5	10.30	10.28	104.1	176	106	331	225
28-100-300-25	4- 9-13	139	104	29.1	1.95	42.5	45.8	10.14	10.36	104.2	176	104	323	219
29-100-200-25	3-20-13	138	105	29.0	2.52	44.3	47.2	10.58	10.68	104.3	169	108	213	105
30-100-200-25	4- 9-13	124	105	29.1	2.32	42.7	45.9	10.22	10.39	104.3	171	96	209	113
31-100-100-25	4- 3-13	135	105	28.9	2.97	41.6	45.3	9.92	10.22	104.1	166	....	....	....
32-80-400-25	3-15-13	141	84	28.8	1.38	33.6	35.7	8.03	8.09	104.4	136	102	448	346
33-80-400-25	4- 9-13	126	83	29.1	1.28	32.2	34.8	7.68	7.86	104.1	130	101	437	336
34-80-300-25	3-19-13	109	83	29.1	1.59	33.1	35.6	7.88	8.03	104.0	132	92	334	242
35-80-300-25	4- 7-13	114	83	29.4	1.55	32.1	34.2	7.65	7.72	104.1	131	84	321	237
36-80-200-25	3-20-13	125	84	29.0	1.92	32.8	34.8	7.83	7.87	104.2	119	102	215	113
37-80-200-25	4- 7-13	103	84	29.4	1.84	32.6	34.4	7.77	7.77	104.1	131	81	218	137
38-80-100-25	3-26-13	139	84	29.2	2.31	31.0	32.6	7.41	7.38	104.4	122	....	....	....
39-80-100-25	4- 3-13	129	84	28.9	2.34	32.0	34.8	7.62	7.85	104.0	122	....	....	....
40-60-400-25	3-19-13	95	62	29.3	0.95	23.3	23.9	5.55	5.40	104.1	82	83	458	375
41-60-400-25	3-28-13	106	62	29.5	0.96	22.7	23.2	5.41	5.40	104.1	93	84	452	368
42-60-300-25	3-19-13	101	62	29.3	1.06	22.6	23.6	5.29	5.33	104.1	82	87	327	240
43-60-300-25	3-28-13	103	63	29.5	1.07	22.8	23.8	5.43	5.38	104.1	93	83	347	264
44-60-200-25	3-28-13	115	63	29.5	1.33	22.9	24.0	5.46	5.42	104.0	90	81	206	125
45-60-200-25	4- 7-13	96	63	29.4	1.31	22.9	23.7	5.47	5.36	104.3	80	80	220	140
46-60-100-25	3-26-13	132	63	29.2	1.59	22.0	23.9	5.25	5.41	104.3	72	....	....	....
47-60-100-25	4- 3-13	122	63	28.9	1.60	22.6	24.2	5.39	5.47	104.1	78	....	....	....

TABLE 6 (CONTINUED)

OBSERVED VALUES FOR ENGINE TESTS WITH REHEATED AIR

Designation		Venturi Meter				Indicator Data				b. h. p. Data		Reheater		
	1	2	3	4	5	6	7	8	9	10	11	12	13	14
	Date	Temperature of Air Upstream (deg. F.)	Pressure of Air Upstream (lb. per sp. in. Gage)	Barometer (in. Mercury)	Venturi Meter Manometer (in. Mercury)	m. e. p. Head End of Cylinder	m. e. p. Crank End of Cylinder	i. h. p. Head End of Cylinder	i. h. p. Crank End of Cylinder	Revolutions per min.	Weight on Scales	Temperature Entering Reheater (deg. F.)	Temperature Leaving Reheater (deg. F.)	Rise of Temperature (deg. F.)
48–100–400–25	5–14–13	138	104	29.0	1.55	43.0	45.5	10.26	10.29	104.2	181	116	432	316
49–100–300–25	5–15–13	138	105	29.0	1.81	42.2	45.0	10.07	10.17	104.2	176	116	313	197
50–100–200–25	5–20–13	127	104	29.0	2.18	43.9	44.8	10.45	10.11	104.0	171	107	216	109
51–100–100–25	5–28–13	137	105	29.1	2.80	42.0	45.2	9.98	10.18	103.8	166	....	....	....
52–80–400–25	5–19–13	116	83	29.3	1.14	32.7	34.1	7.80	7.71	104.2	133	96	433	337
53–80–300–25	5–19–13	110	84	29.3	1.36	32.6	34.1	7.77	7.70	104.1	133	97	324	227
54–80–200–25	5–20–13	119	84	29.1	1.67	33.0	33.5	7.86	7.56	104.1	131	99	215	116
55–80–100–25	5–28–13	115	84	29.1	2.18	32.2	35.6	7.67	8.03	104.0	114	....	....	....
56–60–400–25	5–19–13	110	62	29.3	0.81	22.5	23.7	5.37	5.36	104.2	86	93	445	352
57–60–300–25	5–19–13	110	63	29.3	1.03	23.2	24.3	5.53	5.48	104.0	83	94	330	236
58–60–200–25	5–20–13	106	63	29.1	1.25	23.7	24.7	5.64	5.58	104.0	82	93	216	123
59–60–100–25	5–28–13	120	64	29.1	1.58	23.2	25.3	5.53	5.71	104.0	72	....	....	....
60–100–400–15	5–21–13	126	103	29.0	0.76	29.3	30.6	7.07	6.99	105.3	120	114	439	325
61–100–400–20	5–21–13	135	104	29.0	1.11	36.9	39.1	8.85	8.88	104.7	152	118	427	309
62–100–400–30	5–21–13	145	105	29.0	1.97	49.7	51.0	11.82	11.50	103.9	206	122	425	303
63–100–400–35	5–21–13	147	107	29.0	2.56	56.7	56.3	13.41	12.62	103.3	236	124	436	312

23 to 47, inclusive, are the first tests with incomplete expansion; and those numbered 60 to 63, inclusive, are later tests with variable cut-off. Obviously this classification is overlapping. Thus the variable cut-off tests are also incomplete expansion tests. The object in view in tests 17 to 22, inclusive, was the determination of the effects of cut-off when the temperature and pressure were maintained constant at 400 degrees Fahrenheit and 100 pounds per square inch gage, respectively.

16. *Complete Expansion Tests.*—In these tests the load was adjusted to give a cut-off such that the release was at atmospheric pressure and substantially at the end of the stroke; in other words the "toe" of the indicator diagram came to a well defined point and was never rounded or looped. As shown by the designation, these tests were run at four different gage pressures, including 40, 60, 80, and 100 pounds per square inch. Each pressure was combined with four different temperatures, including 100, 200, 300, and 400 degrees Fahrenheit.

It was in the complete expansion tests that freezing of the exhaust first caused trouble. To expand from 100 pounds gage to atmospheric pressure requires a ratio of expansion of about 5.3. The cut-off at this pressure is shown by the indicator diagrams to be at about 15 per cent of the stroke, the release pressure about 1 pound above atmosphere, and the drop of pressure at cut-off about 12 per cent of the initial gage pressure. Then, if subscript 1 denotes conditions at cut-off and subscript 2 those at release, we have the formula:

$$\frac{p_1}{p_2} = \frac{(v_2)^n}{v_1} = \left(\frac{100 + \text{clearance per cent}}{\text{cut-off per cent} + \text{clearance per cent}}\right)^n$$

Taking atmospheric pressure at 14.5 pounds, which was about the average condition for the tests, we have:

$$p_1 = 100 - 12 + 14.5 = 102.5$$
$$p_2 = 14.5 + 1 = 15.5$$

$$\frac{102.5}{15.5} = \left(\frac{105.5}{20.5}\right)^n \text{ or } 6.61 = 5.15^n$$

$$n = \frac{\log. 6.61}{\log. 5.15} = \frac{0.821}{0.713} = 1.15$$

which checks very well with the value of n obtained by plotting the indicator diagrams taken under these conditions on logarithmic cross-section paper. This value of n shows a considerable heat exchange between the working medium and the cylinder walls. From the formula

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{n-1} = 5.15^{0.15}$$

we find that for air at 100 degrees at cut-off $T_1 = 560$ and $T_2 = 438$, that is, the release temperature was −22 degrees Fahrenheit. It is probable, however, that at cut-off the temperature of the air was at least 20 degrees below the throttle temperature, as the drop in temperature between the throttle and steam chest is 5 degrees for tests of this kind. This would make the temperature at release 37 degrees below zero Fahrenheit. It is not to be inferred that the value of n found above is constant under all conditions. With cut-off at 25 per cent of the stroke, or more, the average value is about $n = 1.275$. In test 4–100–100–15 the temperature registered by the thermometer in the exhaust pipe was −11 degrees Fahrenheit. With a later cut-off and consequently larger amount of air this same thermometer registered 10 to 15 degrees lower than this (see test 31–100–100–25). It might be expected that the lowest temperature would be obtained with earliest cut-off. This would occur if the value of n remained constant under all conditions of operation, but with quantities of air as small as those admitted when the cut-off is at only 15 per cent of the stroke the heat absorbed during the stroke has a greater relative warming effect. In consequence the value of n increases; conversely, the expansion becomes more nearly adiabatic as the cut-off is made later.

Calculating the release temperature when $n = 1.275$, the formula becomes:

$$\frac{T_1}{T_2} = \left(\frac{100 + 5.5}{25 + 5.5}\right)^{0.275}$$

For $T_1 = 560$, this gives $T_2 = 398$, or 62 degrees below zero Fahrenheit. The fall of temperature between throttle and steam chest is not so great in this case as with the 15 per cent cut-off. We may therefore assume that the temperature at cut-off is nearer the throttle temperature but somewhat below it. It is safe to say that the temperature of the air in the cylinder just as the valve is on the point of releasing is between 35 and 50 degrees below zero when the cut-off is at from 15 to 20 per cent of the stroke, and 60 to 70 degrees below zero when the cut-off is at 20 to 25 per cent of the stroke, the temperature at the throttle being in all cases 100 degrees Fahrenheit. After reaching the value 1.275 at 25 per cent cut-off n changes very little, so that the release temperature now depends only on the ratio of expansion, and as this diminishes with later cut-off the temperature in the exhaust again rises as the cut-off becomes later than 25 per cent.

With such temperatures prevailing in the cylinder and exhaust pipe, any moisture in the air is certain to cause operating difficulties in a few minutes. The exhaust ports and exhaust pipe become coated with ice, resulting in an increase in the back pressure which may in time prevent operation of the engine. During the tests, vigorous pounding of the exhaust pipe was frequently necessary to clear it of ice. This was done as soon as the indicator diagrams gave evidence, through an increase in the back pressure, of such ice accumulation.

In addition to the trouble from the formation of ice, the low temperature in the cylinder seriously interfered with effective lubrication. The increased viscosity of the lubricant resulting from the low temperature in the cylinder caused a large internal loss by friction. This difficulty was particularly pronounced when a mixture of cylinder oil and graphite was used as the cylinder lubricant. These two effects of the cold caused the brake horse power and mechanical efficiency of the engine to show considerable irregularities.

Various expedients were adopted to eliminate the lubrication difficulties but without complete success in any instance. Figure 11 shows the effect of variable lubrication produced through the infrequent injection of oil into the cylinder. When oil was thus injected it was noticed that a surprisingly large addition of weight was required to keep the brake scales in equilibrium immediately after the injection. This method of feeding oil was therefore immediately abandoned. In Figure 11 the heavy line is the indicator diagram taken just before lubrication; the dotted line is that taken after lubrication. The interval between the two diagrams was less than 10 minutes. No lubricant had been injected for more than one hour prior to taking the first of these diagrams.

At times the combustion in the reheater was extremely imperfect, in which case a third factor affecting the mechanical efficiency was introduced, one whose effect was entirely unmeasurable. When large quantities of unburned carbon with possibly some free gasoline passed into the cylinder, it is probable that the cylinder lubricant became more viscous because of the presence of the carbon, or its lubricating qualities may have been partially destroyed by the presence of free gasoline. To reduce these lubrication difficulties, the engine was finally equipped with a force feed lubricator set to feed 9 drops per

TABLE 7

OBSERVED VALUES AND COMPUTED RESULTS FOR ENGINE TESTS WITH REHEATED AIR

Designation	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
			Total Air					Air Rate					Temperature (deg. F.)				
	Pressure at Throttle (lb. per sq. in. Gage)	Temp. at Throttle (deg. F.)	lb. per hr.	cu. ft. Free Air per min. (60° F., 14.7 lb. per sq. in. Abs.)	i. h. p.	b. h. p.	Mech. Efficiency	lb. per hr. per i. h. p.	cu. ft. Free Air per min. per i. h. p. (62° F., 14.7 lb. per sq. in. Abs.)	lb. per hr. per b. h. p.	cu. ft. Free Air per min. per b. h. p. (62° F., 14.7 lb. per sq. in., Abs.)	Average Cut-off Per Cent of Stroke	In Steam Chest	At Exhaust	Room	Barometer (in. Mercury)	B. t. u. Req. to Raise Air (Item 3) from 100° F. to Throttle Temp.
1-100-400-15	100	400	505	111	13.66	12.31	90.2	37.1	8.14	41.1	9.00	14.2	336	62	78	29.0	36 400
2-100-300-15	100	300	527	114	13.38	11.78	88.1	38.0	8.35	44.8	9.81	14.8	269	54	77	29.0	25 340
3-100-200-15	100	200	574	126	13.54	11.25	83.2	42.3	9.27	49.7	10.87	14.5	188	34	76	29.0	13 760
4-100-100-15	100	100	684	150	14.11	10.69	75.7	48.5	10.61	64.0	14.03	15.8	95	—11	76	29.2	0
5- 80-400-20	80	400	481	106	12.66	11.17	88.2	38.0	8.35	43.0	9.44	18.6	297	65	73	29.4	34 600
6- 80-300-20	80	300	520	115	12.51	10.65	85.1	41.6	9.13	48.8	10.70	18.6	225	39	71	29.4	34 960
7- 80-200-20	80	200	522	115	12.35	9.10	73.7	42.3	9.27	57.4	12.57	17.5	169	15	64	29.0	.....
8- 80-100-20	80	100	680	149	13.15	9.48	72.0	51.7	11.33	71.7	15.70	21.1	92	— 9	73	29.2	.....
9- 60-400-25	60	400	446	99	10.80	9.49	87.1	40.9	8.98	47.0	10.28	24.8	298	83	73	29.4	32 340
10- 60-300-25	60	300	481	106	10.75	9.00	83.7	44.7	9.80	53.5	11.71	23.8	222	54	70	28.9	23 080
11- 60-200-25	60	200	522	115	10.90	8.50	78.0	47.9	10.50	61.5	13.46	24.1	169	37	67	29.0	12 520
12- 60-100-25	60	100	614	135	10.84	7.44	68.6	56.6	12.41	82.5	18.05	24.8	91	7	80	29.2	0
13- 40-400-35	40	400	416	92	8.81	7.47	84.8	47.2	10.32	55.5	12.20	34.7	301	104	74	29.4	40 280
14- 40-300-35	40	300	435	96	8.35	6.65	79.6	52.1	11.43	65.4	14.33	32.7	215	64	69	28.9	20 880
15- 40-200-35	40	200	480	104	8.71	5.83	67.0	55.1	12.08	82.4	18.05	34.0	166	48	68	29.0	11 520
16- 40-100-35	40	100	524	116	8.36	5.00	59.8	62.7	13.75	104.8	22.97	32.5	95	25	80	29.2	0
17-100-400-15	100	400	514	114	14.27	12.98	91.0	36.7	8.02	40.6	8.82	15.7	356	66	69	29.3	37 700
18-100-400-17.5	100	400	561	122	15.67	14.37	91.7	35.9	7.87	39.0	8.55	17.4	340	66	78	29.3	40 380
19-100-400-20	100	400	622	136	17.67	16.09	91.1	35.2	7.73	38.6	8.44	20.0	360	64	78	28.8	44 800
20-100-400-20	100	400	626	137	18.22	16.42	90.1	34.4	7.54	38.1	8.35	20.3	342	64	76	29.3	45 050
21-100-400-25	100	400	715	157	20.98	18.96	90.3	34.1	7.48	37.7	8.26	24.8	344	62	75	29.4	51 500
22-100-400-25	100	400	715	157	20.81	18.87	90.6	34.3	7.52	37.9	8.30	24.8	363	65	73	28.8	51 500
23-100-400-27.5	100	400	772	169	22.63	20.88	92.2	34.1	7.48	37.0	8.10	27.7	346	56	70	29.4	55 620
24-100-400-30	100	400	812	178	23.34	21.73	93.1	34.8	7.63	37.4	8.19	29.7	364	62	71	29.3	58 470
25-100-400-30	100	400	802	176	23.61	21.43	90.7	34.0	7.45	37.5	8.21	29.7	367	64	68	28 8	57 800
26-100-400-35	100	400	913	200	26.16	24.47	93.5	34.9	7.63	37.3	8.16	34.4	370	67	69	29.3	65 700
22-100-400-25	100	400	715	157	20.81	18.87	90.6	34.3	7.52	37.9	8.30	24.8	363	65	73	28.8	51 500
27-100-300-25	100	300	775	170	20.58	18.32	89.1	37.6	8.24	42.2	9.23	24.6	268	45	77	29.1	37 220
28-100-300-25	100	300	763	168	20.50	18.33	89.4	37.3	8.18	41.6	9.10	25.5	266	35	66	29.1	36 650
29-100-200-25	100	200	866	189	21.26	17.63	82.9	40.7	8.92	49.0	10.73	25.0	181	14	66	29.0	20 780
30-100-200-25	100	200	843	184	20.61	17.84	86.5	40.9	8.96	47.2	10.32	25.2	184	0	66	29.1	20 240
31-100-100-25	100	100	937	205	20.14	17.29	85.4	46.6	10.20	54.2	11.86	25.5	100	—26	82	28.9	0
32- 80-400-25	80	400	586	129	16.12	14.19	88.0	36.4	7.97	41.3	9.04	25.4	357	73	74	28.8	42 200
33- 80-400-25	80	400	573	126	15.54	13.53	87.2	36.8	8.06	42.3	9.27	25.3	342	50	73	29.1	41 230
34- 80-300-25	80	300	645	141	15.91	13.73	86.3	40.6	8.90	46.9	10.29	25.1	263	51	78	29.1	30 960
35- 80-300-25	80	300	643	144	15.37	13.04	88.7	41.2	9.05	46.4	10.18	24.6	254	36	70	29.4	30 390
36- 80-200-25	80	200	699	154	15.70	12.40	79.0	44.5	9.76	56.4	12.36	24.9	180	22	70	29.0	16 790
37- 80-200-25	80	200	696	152	15.54	13.64	87.8	44.8	9.83	51.0	11.16	25.1	185	17	68	29.4	16 680
38- 80-100-25	80	100	757	166	14.79	12 74	86.2	51.2	11.21	59.3	12.99	24.2	89	—38	70	29.2	0
39- 80-100-25	80	100	765	168	15.47	12.69	82.0	49.4	10.82	60.4	13.21	25.1	95	—12	82	28.9	0
40- 60-400-25	60	400	451	99	10.95	8.54	78.0	41.2	9.05	52.8	11.56	24.9	321	81	73	29.3	32 500
41- 60-400-25	60	400	450	99	10.81	9.68	89.5	41.6	9.13	46.4	10.18	25.2	338	74	71	29.5	32 450
42- 60-300-25	60	300	473	104	10.72	8.54	79.6	44.1	9.69	55.4	12.13	24.8	260	66	77	29.3	22 700
43- 60-300-25	60	300	477	104	10.81	9.68	89.5	44.0	9.67	49.2	10.77	25.1	269	64	68	29.5	22 900
44- 60-200-25	60	200	536	115	10.88	9.36	86.0	48.2	10.56	56.1	12.29	24.9	185	37	62	29.5	12 610
45- 60-200-25	60	200	540	116	10.83	8.34	76.9	48.9	10.72	63.4	13.90	24.1	182	32	69	29.4	12 700
46- 60-100-25	60	100	566	124	10.66	7.51	70.5	53.1	11.61	75.4	16.51	25.3	88	—10	59	29.2	0
47- 60-100-25	60	100	573	126	10.86	8.12	74.7	52.8	11.57	70.5	15.45	24.2	97	10	80	28.9	0
48-100-400-25	100	400	683	149	20.55	18.87	91.8	33.3	7.30	36.2	7.91	23.9	344	65	93	29.0	49 200
49-100-300-25	100	300	738	162	20.24	18.33	90.6	36.5	8.01	40.2	8.80	24.5	268	51	91	29.0	35 440
50-100-200-25	100	200	816	178	20.56	17.79	86.5	39.7	8.69	45.8	10.05	24.6	187	23	89	29.0	19 580
51-100-100-25	100	100	914	200	20.16	15.69	77.8	45.3	9.94	58.3	12.78	24.4	98	—24	85	29.1	0
52- 80-400-25	80	400	546	119	15.51	13.86	89.3	35.2	7.73	39.5	8.64	23.8	337	66	81	29.3	39 300
53- 80-300-25	80	300	598	131	15.47	13.85	89.4	38.6	8.47	43.2	9.46	23.8	267	54	82	29.3	28 690
54- 80-200-25	80	200	657	144	15.42	12.90	83.7	42.5	9.33	50.8	11.11	24.0	184	27	86	29.1	15 710
55- 80-100-25	80	100	752	165	15.70	11.86	75.5	47.9	10.48	63.3	13.88	24.7	....	— 4	90	29.1	0
56- 60-400-25	60	400	411	90	10.73	8.96	83.5	38.3	8.40	45.8	10.05	24.1	321	83	85	29.3	29 610
57- 60-300-25	60	300	464	102	11.01	8.63	78.4	42.0	9.22	53.8	11.78	25.0	259	64	84	29.3	22 250
58- 60-200-25	60	200	514	112	11.27	8.52	76.0	45.7	10.70	60.3	13.21	24.4	180	39	84	29.1	12 300
59- 60-100-25	60	100	573	126	11.24	7.49	66.1	51.0	11.16	76.4	16.73	24.4	....	16	90	29.1	0
60-100-400-15	100	400	487	107	14.06	12.64	89.2	34.6	7.59	38.5	8.39	14.1	340	71	87	29.0	35 120
61-100-400-20	100	400	586	128	17.73	15.92	89.8	32.9	7.21	36.7	8.03	19.2	347	70	87	29.0	42 040
48-100-400-25	100	400	683	149	20.55	18.87	91.8	33.3	7.30	36.2	7.91	23.9	344	65	93	29.0	49 200
62-100-400-30	100	400	765	168	23.32	21.40	91.8	32.9	7.21	35.9	7.86	28.7	355	71	87	29.0	55 200
63-100-400-35	100	400	874	192	26.03	24.40	93.7	33.6	7.36	35.9	7.86	33.5	357	71	87	29.0	62 850

(Table 7, continued)

Designation	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34
														13 000 B. t. u. Fuel Applied at 60% Efficiency		20 000 B. t. u. Fuel Applied at 80% Efficiency	
	lb. Air Req. per i. h. p. hr. at 100° F. and Pressure in Item 1	i. h. p. hr. Obtainable from Air (Item 3) at 100° F. and Pressure in Item 1	i. h. p. hr. Gained (Item 5 Less Item 19)	B. t. u. Req. per i. h. p. hr. Gained (Item 17 + Item 20)	Thermal Efficiency (2546 ÷ Item 21)	Ideal Efficiency (See Note)	Efficiency Ratio (Item 22 ÷ Item 23)	lb. Air Req. per b. h. p. hr. at 100° F. and Pressure in Item 1	b. h. p. hr. Obtainable from Air (Item 3) at 100° F. and Pressure in Item 1	b. h. p. hr. Gained (Item 6 Less Item 26)	B. t. u. Req. per b. h. p. hr. Gained (Item 17 + Item 27)	i. h. p. hr. Gained per 1000 B. t. u. Added	b. h. p. hr. Gained per 1000 B. t. u. Added	lb. per hr. per i. h. p. hr. Gained	lb. per hr. per b. h. p. hr. Gained	lb. per hr. per i. h. p. hr. Gained	lb. per hr. per b. h. p. hr. Gained
1-100-400-15	48.3	10.46	3.20	11 350	22.4	44.4	50.6	60.8	8.31	4.00	9 110	0.088	0.111	1.45	1.16	0.71	0.57
2-100-300-15	48.3	10.91	2.47	10 250	24.9	44.4	56.0	60.8	8.68	3.10	8 170	0.098	0.122	1.31	1.05	0.64	0.52
3-100-200-15	48.3	11.89	1.65	8 370	30.4	44.4	68.5	60.8	9.44	1.81	7 020	0.118	0.142	1.07	0.91	0.53	0.44
4-100-100-15	48.3	.....	....	.....	....	....	....	....	....	....	....	....	....	....	....	....	....
5- 80-400-20	51.7	9.31	3.35	10 310	24.7	41.3	69.7	68.0	7.08	4.09	8 470	0.097	0.118	1.32	1.08	0.65	0.53
6- 80-300-20	51.7	10.03	2.48	10 100	25.2	41.3	61.1	68.0	7.65	3.00	8 320	0.099	0.120	1.29	1.07	0.63	0.52
7- 80-200-20	.....	.....	....	.....	....	....	....	....	....	....	....	....	....	....	....	....	....
8- 80-100-20	....	.....	....	.....	....	....	....	....	....	....	....	....	....	....	....	....	....
9- 60-400-25	54.4	8.22	2.67	12 080	21.1	37.1	56.9	73.1	6.10	3.39	9 440	0.082	0.105	1.55	1.23	0.75	0.60
10- 60-300-25	54.4	8.84	1.91	12 070	21.1	37.1	56.9	73.1	6.58	2.42	9 440	0.082	0.105	1.55	1.23	0.75	0.60
11- 60-200-25	54.4	9.61	1.29	9 730	26.1	37.1	70.6	73.1	7.05	1.45	8 640	0.103	0.116	1.25	1.11	0.61	0.54
12- 60-100-25	54.4	.....	....	.....	....	....	....	....	....	....	....	....	....	....	....	....	....
13- 40-400-35	61.8	6.73	2.08	19 350	13.2	31.3	41.3	104.8	3.97	3.50	11 500	0.051	0.087	2.48	1.47	1.21	0.72
14- 40-300-35	61.8	7.03	1.32	15 800	16.1	31.3	51.6	104.8	4.15	2.50	8 360	0.063	0.119	2.02	1.07	0.99	0.53
15- 40-200-35	61.8	7.77	0.94	12 250	20.8	31.3	66.4	104.8	4.58	1.25	9 210	0.082	0.109	1.57	1.18	0.76	0.58
16- 40-100-35	61.8	.....	....	.....	....	....	....	....	....	....	....	....	....	....	....	....	....
17-100-400-15	48.2	10.83	3.44	10 920	23.3	44.4	53.6	60.8	8.61	4.37	8 630	0.091	0.116	1.40	1.10	0.63	0.55
18-100-400-17.5	47.6	11.77	3.90	10 330	24.7	44.4	55.6	58.7	9.56	4.18	9 650	0.097	0.104	1.33	1.24	0.65	0.61
19-100-400-20	47.4	13.17	4.50	9 660	25.5	44.4	57.6	56.9	10.94	5.15	9 700	0.100	0.115	1.28	1.11	0.62	0.55
20-100-400-20	47.4	13.25	4.97	9 080	28.1	44.4	63.3	56.9	11.04	5.38	9 150	0.111	0.110	1.16	1.17	0.57	0.58
21-100-400-23	46.6	15.33	5.65	9 110	28.0	44.4	63.1	54.4	13.15	5.81	8 880	0.110	0.113	1.16	1.13	0.57	0.56
22-100-400-25	46.6	15.33	5.48	9 390	27.1	44.4	61.2	54.4	13.15	5.73	8 990	0.107	0.112	1.21	1.15	0.59	0.57
23-100-400-27.5	46.6	16.57	6.06	9 170	27.8	44.4	62.6	54.4	14.18	6.70	8 300	0.109	0.120	1.17	1.06	0.58	0.52
24-100-400-30	46.6	17.43	5.91	9 900	25.7	44.4	58.0	54.4	14.91	6.82	8 570	0.101	0.116	1.27	1.10	0.62	0.54
25-100-400-30	46.6	17.23	6.38	9 060	28.2	44.4	63.5	54.4	14.75	6.08	8 650	0.111	0.115	1.15	1.11	0.57	0.55
26-100-400-35	46.6	19.56	6.60	9 960	25.6	44.4	57.9	54.4	16.77	7.70	8 540	0.109	0.117	1.28	1.09	0.62	0.54
22-100-400-23	46.6	15.33	5.48	9 390	27.1	44.4	61.2	54.4	13.15	5.73	8 990	0.107	0.112	1.21	1.15	0.59	0.57
27-100-300-25	46.6	16.65	3.93	9 725	26.1	44.4	59.0	54.4	14.23	4.09	9 110	0.103	0.109	1.25	1.16	0.61	0.57
28-100-300-25	46.6	16.38	4.12	8 900	28.7	44.4	64.6	54.4	14.03	4.30	8 530	0.113	0.117	1.13	1.09	0.56	0.54
29-100-200-25	46.6	18.58	2.68	7 755	32.8	44.4	74.1	54.4	15.90	1.73	12 000	0.129	0.084	1.00	1.55	0.48	0.75
30-100-200-25	46.6	18.10	2.51	8 075	31.6	44.4	71.2	54.4	15.49	2.35	8 630	0.124	0.116	1.03	1.10	0.50	0.54
31-100-100-25	46.6	.....	....	.....	....	....	....	....	....	....	....	....	....	....	....	....	....
32- 80-400-25	50.5	11.60	4.52	9 330	27.3	41.3	66.0	60.8	9.65	4.54	9 300	0.107	0.108	1.20	1.20	0.59	0.58
33- 80-400-25	50.5	11.33	4.21	9 800	26.0	41.3	63.1	60.8	9.43	4.10	10 060	0.102	0.099	1.26	1.29	0.61	0.63
34- 80-300-25	50.5	12.77	3.14	9 860	25.8	41.3	62.6	60.8	10.60	3.13	9 900	0.101	0.101	1.27	1.27	0.62	0.60
35- 80-300-25	50.5	12.52	2.85	10 660	23.9	41.3	57.9	60.8	10.42	3.22	9 440	0.094	0.106	1.37	1.22	0.67	0.59
36- 80-200-25	50.5	13.84	1.86	9 025	28.2	41.3	68.3	60.8	11.51	0.89	18 850	0.111	0.053	1.14	2.42	0.57	1.17
37- 80-200-25	50.5	13.79	1.75	9 550	26.6	41.3	64.5	60.8	11.45	2.19	7 620	0.105	0.131	1.23	0.98	0.60	0.47
38- 80-100-25	50.5	.....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
39- 80-100-25	50.5	.....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
40- 60-400-25	54.4	8.31	2.64	12 300	20.7	37.1	56.0	73.2	6.17	2.37	13 700	0.082	0.073	1.58	1.75	0.77	0.86
41- 60-400-25	54.4	8.28	2.53	12 810	19.8	37.1	53.5	73.2	6.16	3.52	9 220	0.078	0.109	1.64	1.18	0.80	0.58
42- 60-300-25	54.4	8.69	2.03	11 180	22.7	37.1	61.2	73.2	6.46	2.08	10 920	0.089	0.091	1.43	1.40	0.70	0.68
43- 60-300-25	54.4	8.79	2.02	11 340	22.4	37.1	60.4	73.2	6.52	3.16	7 250	0.088	0.139	1.45	0.93	0.71	0.45
44- 60-200-25	54.4	9.66	1.22	10 320	24.7	37.1	66.6	73.2	7.18	2.18	5 780	0.097	0.173	1.32	0.74	0.65	0.36
45- 60-200-25	54.4	9.73	1.10	11 550	22.0	37.1	59.5	73.2	7.24	1.10	11 530	0.086	0.086	1.48	1.48	0.72	0.72
46- 60-100-25	54.4	.....	....	.....	....	....	....	....	....	....	....	....	....	....	....	....	....
47- 60-100-25	54.4	.....	....	.....	....	....	....	....	....	....	....	....	....	....	....	....	....
48-100-400-25	45.3	15.04	5.51	8 920	28.6	44.4	64.4	54.4	12.55	6.32	7 790	0.112	0.128	1.14	1.00	0.56	0.49
49-100-300-25	45.3	16.30	3.94	8 990	28.3	44.4	63.9	54.4	13.55	4.78	7 420	0.111	0.135	1.15	0.95	0.57	0.46
50-100-200-25	45.3	18.00	2.56	7 640	33.3	44.4	75.2	54.4	15.00	2.79	7 010	0.131	0.143	0.98	0.90	0.48	0.44
51-100-100-25	45.3	20.16	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
52- 80-400-25	48.0	11.37	4.14	9 490	26.8	41.3	64.9	60.8	8.98	4.88	9 060	0.105	0.123	1.22	1.03	0.60	0.50
53- 80-300-25	48.0	12.43	3.02	9 510	26.8	41.3	64.9	60.8	9.83	4.02	7 130	0.105	0.141	1.22	0.92	0.60	0.44
54- 80-200-25	48.0	13.67	1.75	8 980	28.3	41.3	68.6	60.8	10.80	2.10	7 480	0.111	0.134	1.15	0.98	0.56	0.46
55- 80-100-25	48.0	15.70	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
56- 60-400-25	51.1	8.05	2.68	11 030	23.1	37.1	62.4	73.1	5.62	3.34	8 880	0.090	0.113	1.41	1.14	0.69	0.56
57- 60-300-25	51.1	9.08	1.93	11 500	22.1	37.1	59.8	73.1	6.34	2.29	9 730	0.087	0.103	1.47	1.25	0.72	0.61
58- 60-200-25	51.1	10.05	1.17	10 500	24.3	37.1	65.5	73.1	7.02	1.50	8 200	0.095	0.122	1.35	1.05	0.66	0.52
59- 60-100-25	51.1	11.24	....	.....	....	....	....	....	....	....	....	....	....	....	....	....	....
60-100-400-15	43.8	10.45	3.61	9 740	26.1	44.4	58.8	60.8	8.02	4.62	7 600	0.103	0.132	1.25	0.97	0.61	0.47
61-100-400-20	45.8	12.75	4.98	8 460	30.1	44.4	67.9	56.9	10.28	5.64	7 480	0.122	0.134	1.08	0.96	0.53	0.46
48-100-400-25	45.3	15.04	5.51	8 920	28.6	44.4	64.4	54.4	12.55	6.32	7 790	0.112	0.128	1.14	1.00	0.56	0.49
62-100-400-30	45.3	16.90	6.42	8 590	29.6	44.4	66.7	54.4	14.10	7.30	7 580	0.116	0.132	1.10	0.97	0.54	0.47
63-100-400-35	45.3	19.25	6.78	9 260	27.4	44.4	61.8	54.4	16.06	8.34	7 540	0.108	0.133	1.18	0.97	0.58	0.47

NOTE:
Item 23, Ideal Efficiency $= 1 - \left[\frac{14.7}{(\text{Item 1}) + 14.7}\right]^{0.286}$

17. *Variable Cut-off Tests.*—Tests 17 to 22 and 60 to 63 inclusive were made for the purpose of determining the air economy of the engine with different ratios of expansion, other conditions remaining constant. All these tests were therefore made with the same pressure and temperature conditions at the throttle, namely 100 pounds per square inch gage pressure and 400 degrees Fahrenheit. The cut-off was increased from test to test by about 5 per cent, starting with 15 per cent in the first test of each series. At the assigned initial pressure this is the earliest cut-off that could be maintained, since this gives atmospheric pressure at release and any earlier cut-off would have caused a loop of negative work on the diagram.

TABLE 9

EXTREME VARIATIONS OF MECHANICAL EFFICIENCY WITH SIMILAR PRESSURE AND TEMPERATURE CONDITIONS

Cut-off at 25 per cent of stroke

400 deg.		300 deg.		200 deg.		100 deg.	
High	Low	High	Low	High	Low	High	Low
100 lb. Gage							
91.8	90.3	90.6	89.1	86.5	82.9	85.4	77.8
80 lb. Gage							
89.3	87.2	89.4	86.3	87.8	79.0	86.2	75.5
60 lb. Gage							
89.5	78.0	89.5	78.4	86.0	76.0	74.7	66.1

TABLE 10

EXTREME VARIATIONS OF MECHANICAL EFFICIENCY WITH SIMILAR PRESSURE AND TEMPERATURE CONDITIONS

Various Cut-offs*

400 deg.		300 deg.		200 deg.		100 deg.	
High	Low	High	Low	High	Low	High	Low
100 lb. Gage							
93.7 33.5	90.2 14.2	90.6 24.5	88.1 14.8	86.5 25.2	82.9 25.0	85.4 25.5	75.7 15.8
80 lb. Gage							
89.3 23.8	87.2 25.3	89.4 23.8	85.1 18.6	87.8 25.1	73.7 17.5	86.2 24.2	72.0 21.1

TABLE 10 (CONTINUED)

EXTREME VARIATIONS OF MECHANICAL EFFICIENCY WITH SIMILAR PRESSURE AND TEMPERATURE CONDITIONS

Various Cut-offs*

60 lb. Gage							
89.5 25.2	78.0 24.9	89.5 25.1	78.4 25.0	86.0 24.9	76.0 24.4	71.7 24.2	66.1 24.4
40 lb. Gage							
84.8 34.7		79.6 32.7		67.0 31.0		59.8 32.5	

* The lower figures under each pressure are the actual points of cut-off in per cent of the stroke.

18. *Incomplete Expansion Tests.*—Of the 3 tests made on compressed air 34 were incomplete expansion tests, the cut-off in each case being at 25 per cent of the stroke. Since the cut-off was kept constant during these tests the lowest initial pressure which could be maintained without the formation of a loop of negative work on the indicator diagram was 60 pounds per square inch gage.

As shown by Table 7, these tests include three initial pressures, 100, 80, and 60 pounds per square inch gage, each combined with four temperatures at the throttle, 400, 300, 200, and 100 degrees Fahrenheit, respectively. With two exceptions the tests were run under identical conditions; the exceptions being those tests made with a pressure of 100 pounds gage and air temperatures of 400 degrees and 100 degrees Fahrenheit. Tests 27 to 47 inclusive were made during March and April, while tests 48 to 63 inclusive were made during the latter part of May. This repetition of tests under the same conditions seemed desirable when it was observed that there was a perceptible decrease in the air consumption per indicated horse power as the season advanced. The curves of air rate per indicated horse power per hour plotted against temperature for tests made during one month fell slightly below those made during the preceding month in spite of every effort to make all conditions the same. The

May tests gave an air rate per indicated horse power per hour of from 1 to 1½ pounds less than those made in March. Table 8 shows these variations. With a cut-off earlier than 25 per cent the variations in air consumption were even greater than those with 25 per cent cut-off.

Since the engine employed in these tests was new, the gradual reduction in air consumption was largely brought about by improvement in the condition of the cylinder and valves, resulting from wear; and to a lesser extent by the increasing room temperature as the outdoor temperature increased. Thus, during the tests made in May the room temperature was about 20 degrees higher than that maintained in March.

Table 7 presents certain observed and all computed values for the whole series of air-engine tests. While the derivation of the various items is generally obvious, it seems desirable to indicate the source of the data or the method of computation of each item listed.

Items 1 and 2 are average values of data from tests.

Item 3 is computed from the Venturi meter data given in Table 6, using the general method described in the paper mentioned on page 20.

Item 4 = Item 3 × 0.219 (see Section 8).

Item 5 is computed from the engine constants, L = stroke in feet and A = mean area of piston in square inches, the average speed in revolutions per minute, N, and the average mean effective pressure, P, from the various indicator diagrams, by the formula

$$\text{I. h. p.} = \frac{2\ PLAN}{33000}$$

Item 6 is computed from the radius of brake-arm in feet, R, the load on the scales in pounds, W, and the average revolutions per minute, N, by the formula

$$\text{B. h. p} = \frac{2\ \pi\ R\ W\ N}{33000}$$

Item 7 = Item 6 divided by Item 5.
Item 8 = Item 3 divided by Item 5.
Item 9 = Item 4 divided by Item 5.
Item 10 = Item 3 divided by Item 6.
Item 11 = Item 4 divided by Item 6.
Item 12 is obtained from the indicator diagrams.
Items 13, 14, 15, and 16 are average data from the tests.
Item 17 = Item 3 multiplied by the difference between Items 2 and 1 and by the specific heat of air at constant pressure (0.24), the values recorded being the nearest tens.

Item 18 is selected from the curves representing the air consumption per indicated horse power for various throttle temperatures, Figs. 13 and 14, since these values are possibly somewhat more consistent than the results of the individual test at 100 degrees temperature at the throttle; wherefore these values do not agree exactly with the items given under Item 8.

Item 19 = Item 3 divided by Item 18.
Item 20 = Item 5 minus Item 19.
Item 21 = Item 17 divided by Item 20.
Item 22 = the heat equivalent of one horse power per hour (2546 B.t.u.) divided by Item 21.

$$\text{Item 23} = 1 - \left(\frac{14.7}{\text{Item 1} + 14.7}\right)^{0.286}$$

No attempt has been made to determine the ideal efficiency of the tests based upon incomplete expansion of the air. For some of the tests the value of Item 23 is therefore somewhat high.*

Item 24 = Item 22 divided by Item 23. For the reason stated under Item 23, the value of Item 24 is in some cases slightly smaller than would have been the case had the ideal efficiency for incomplete expansion been computed.

Item 25 is obtained from the curves representing the air consumption per brake horse power per hour for the various throttle temperatures. These values were selected for the reason stated under Item 18.

Item 26 = Item 3 divided by Item 25.
Item 27 = Item 6 minus Item 26.
Item 28 = Item 17 divided by Item 27.
Item 29 = Item 20 multiplied by 1000 and divided by Item 17.
Item 30 = Item 27 multiplied by 1000 and divided by Item 17.
Item 31 = 1000 divided by the product of 13 000 and 0.6 and Item 29.
Item 32 = 1000 divided by the product of 13 000 and 0.6 and Item 30.
Item 33 = 1000 divided by the product of 20 000 and 0.8 and Item 29.
Item 34 = 1000 divided by the product of 20 000 and 0.8 and Item 30.

The steam chest temperatures, Item 13, show a surprising drop between the temperature at the throttle and the point in the steam-chest where these temperatures were taken. The amount of drop decreases rapidly with the lowering of the throttle temperature. As

* See Section 5, Case II, for complete adiabatic expansion, which represents the maximum attainable thermal efficiency.

the steam chest thermometer was placed in a thermometer well, it is probable that in the first twenty-one tests these differences are too great. This well was later removed and replaced by a stuffing-box through which the thermometer was passed, thus giving the temperature by direct contact between the bulb and the heated air as it passed to the steam ports of the engine. The drop of temperature is greater the earlier the cut-off, and decreases with later cut-off. For cut-off at 25 per cent of the stroke a difference of 40 to 50 degrees may be expected between throttle and steam chest temperatures when the throttle conditions are 100 pounds gage pressure and 400 degrees Fahrenheit. Exhaust temperatures, Item 14, are particularly unsatisfactory. The thermometer could not be placed near enough to the exhaust port of the engine to give the reading there and the temperature in the exhaust pipe was affected both by the room temperature and the exhaust pipes of other engines.

The heat required to raise the weight of air used in a particular test from 100 degrees Fahrenheit to the temperature maintained at the throttle, Item 17, does not include the loss of heat from the air pipe between the reheater and the throttle. Due to its low specific heat and heat content, the drop in temperature of air flowing in a pipe is large. If the reheater is located l feet from the throttle valve, the temperature, t_r, required at the reheater to maintain a temperature, t_t, at the throttle may be computed by the following equation:

$$\log_e (t_r - t_o) = \frac{\pi D k l}{12 M c_p} + \log_e (t_t - t_o)$$

in which

t_o = temperature of the freezing point, 32 degrees Fahrenheit;
d = diameter of pipe in inches;
l = length of pipe in feet;
m = weight of air flowing per hour;
c_p = specific heat of air at constant pressure, (0.24);
k = coefficient of heat loss in B.t.u. per square foot of surface per degree difference between the air inside the pipe and that outside; approximately 3 for uncovered pipe and 0.6 for covered pipe.

The solution of a specific problem by means of the foregoing equation will show the great importance of locating the reheater as near the engine as possible, and of carefully insulating the connecting pipe and fittings.

The air-rates per indicated horse power per hour for various throttle temperatures are shown in Figs. 13 and 14. The former includes the data from tests 1 to 16 inclusive and 22 to 47 inclusive; the latter, tests 48 to 59 inclusive, made after the engine was well "worn in" and with a room temperature 20 degrees higher than in the earlier tests. The air rate in pounds per indicated horse power per hour for various throttle temperatures and gage pressures is shown in Fig. 15. The air rates per brake horse power per hour for various throttle temperatures are shown in Fig. 16. Some of these data are shown in an even more interesting manner in Fig. 12 in which the various pressure curves are drawn with mechanical efficiencies as ordinates and throttle temperatures as abscissæ.

It was expected that there would be some improvement in the mechanical efficiency with increasing air temperatures, but the extent of this improvement was such as to astonish everyone concerned with the tests. With a throttle temperature of 400 degrees Fahrenheit and

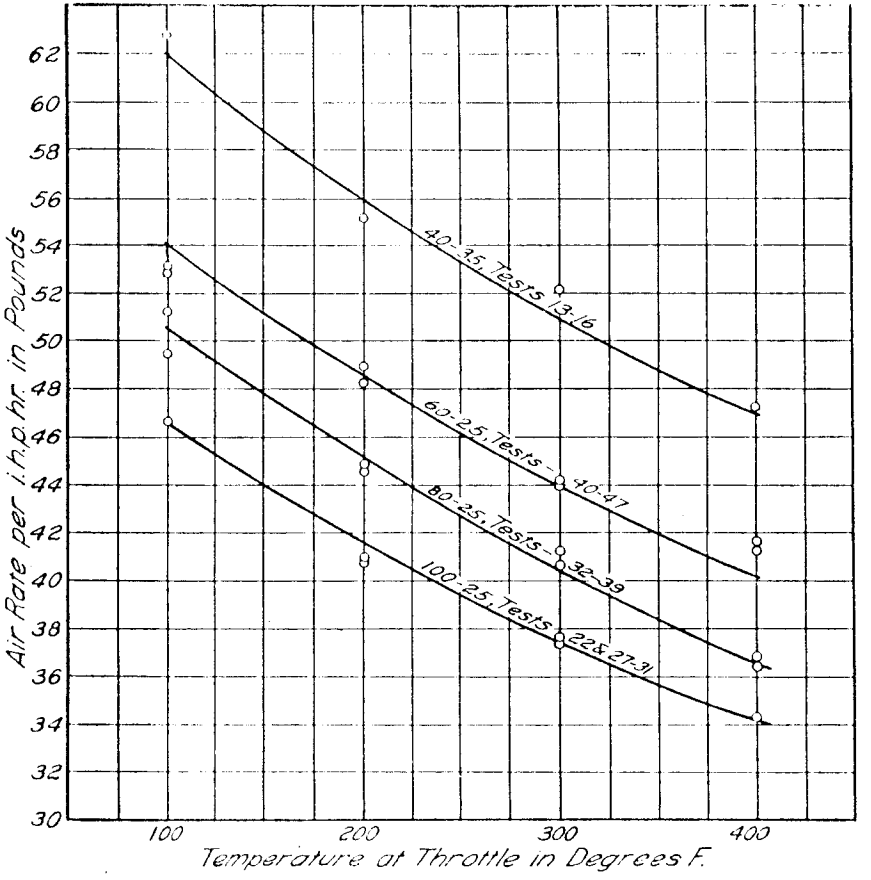

FIG. 13. AIR CONSUMPTION PER INDICATED HORSE POWER PER HOUR FOR VARIOUS THROTTLE TEMPERATURES

a pressure of 100 pounds gage, mechanical efficiencies of 89.2 to 93.7 per cent were obtained; and with a throttle temperature of 100 degrees Fahrenheit and the same pressure, the mechanical efficiencies varied from 75.7 to 85.4 per cent. The lowest recorded mechanical efficiency is that of test 16, its value being only 59.8 per cent. It should be noted that in nearly every test the gain in brake horse power through reheating is greater than the gain in indicated horse power. Tables 9 and 10 present the high and low mechanical efficiencies obtained under various conditions of operation.

Items 31 and 32 (Table 7) present the computed weights of fuel consumed per hour for each indicated and brake horse power gained by reheating. Because of conditions discussed in the preceding paragraph, the weight of fuel per brake horse power per hour is *less* than that required per indicated horse power. While the weight of fuel required per brake horse power per hour fluctuated considerably for different tests, it should be noted that in several tests values of less than one pound of fuel were obtained. Few devices for converting heat into work can show equally satisfactory results.

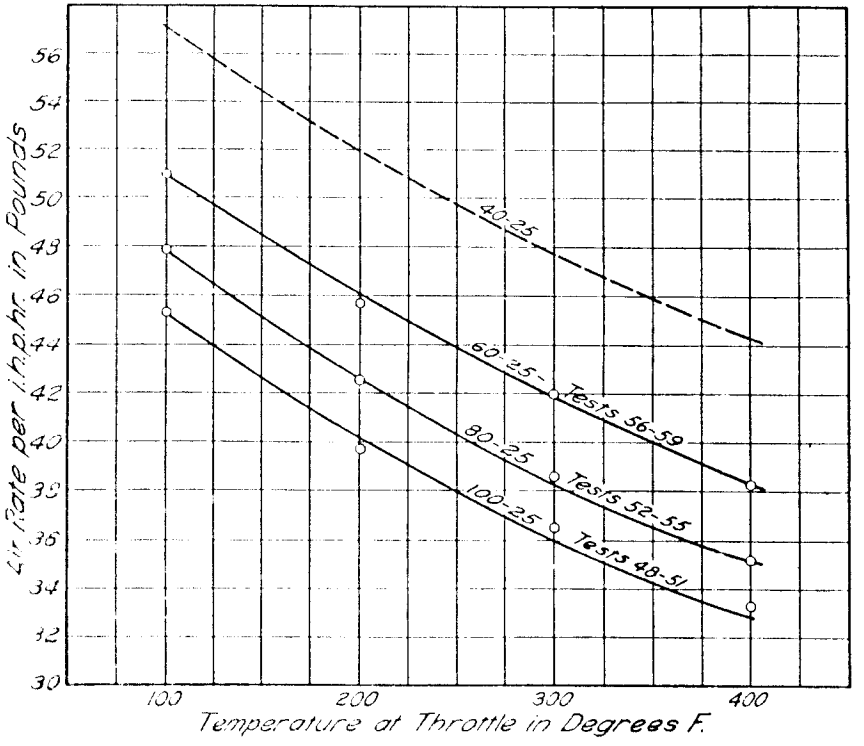

FIG. 14. AIR CONSUMPTION PER INDICATED HORSE POWER PER HOUR FOR VARIOUS THROTTLE TEMPERATURES

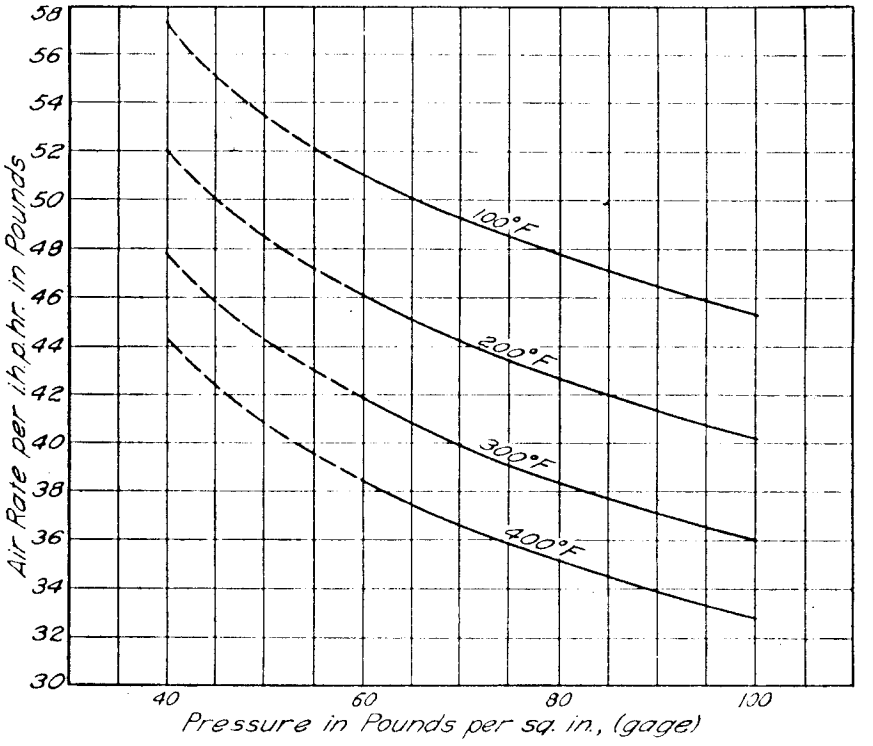

FIG. 15. AIR CONSUMPTION PER INDICATED HORSE POWER PER HOUR FOR VARIOUS THROTTLE TEMPERATURES AND GAGE PRESSURES

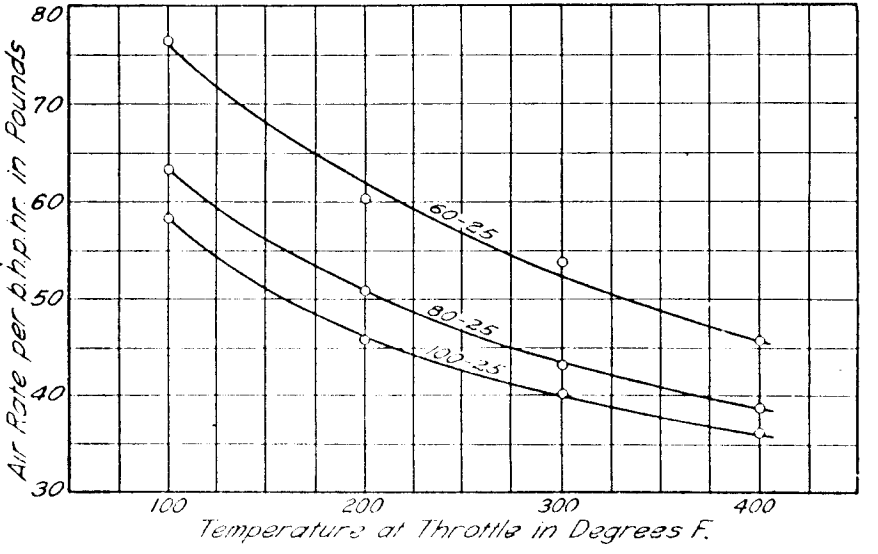

FIG. 16. AIR CONSUMPTION PER BRAKE HORSE POWER PER HOUR FOR VARIOUS THROTTLE TEMPERATURES

V. Tests of Engine Using Mixtures of Air and Steam as the Working Fluid

19. *Purposes of Tests.*—The primary purpose of this series of tests of air-steam mixtures was the determination of the efficiency resulting from the use of steam as a medium for reheating compressed air; the secondary purpose was the devising of methods of testing which might be applicable in a more complete study of the subject, to be undertaken after the present general investigation.

20. *Method of Conducting Tests.*—While the plant described in Section IV was employed in these tests, with the condenser in service, the use of steam necessitated several changes in the methods of testing from those employed with air. The air from the compressor was delivered to the mixing chamber immediately above the throttle valve at a temperature which was maintained as nearly as possible at 100 degrees Fahrenheit; at this point steam was injected into the air until the required conditions of pressure and ratio of air to steam were obtained. To secure these conditions the running of from one to three preliminary tests was necessary. Before making these trial tests, an estimate was made to determine the approximate weight of air and steam required for a given mixture with the engine cut-off at 25 per cent of the stroke. The valve in the air line was regulated until the Venturi meter showed the correct flow of air, when steam was admitted in sufficient quantity to give the required pressure at the throttle. The plant was operated with these valve adjustments until the condenser was discharging steadily and other conditions were constant. When the weight of condensate showed the desired ratio of air and steam the test was started and continued for one hour.

In this test series much less time was required for securing thermal equilibrium in the cylinder and steam chest than in the air tests. The thermometers at the throttle, in the steam chest, and in the exhaust pipe quickly reached definite constant temperatures which were maintained as long as conditions were kept uniform. Thus, despite the fact that considerable time was required for the adjustment of the mixture, the preparatory run was generally no longer than when air alone was used.

The pressure at the throttle and the load on the brake were kept constant by frequent adjustment throughout the test. The following observations were taken as indicated:

(1) Temperature at throttle (every 10 minutes)
(2) Temperature in steam chest (every 10 minutes)
(3) Revolutions per minute (every 10 minutes)
(4) Pressure in mixing chamber (every 10 minutes)
(5) Temperature in mixing chamber (every 10 minutes)
(6) Temperature of air before entering mixing chamber (every 10 minutes)
(7) Temperatures in throttling steam calorimeter (every 10 minutes)
(8) Venturi manometer (every 2 minutes)
(9) Pressure of air before entering mixing chamber (every 10 minutes)
(10) Temperature in exhaust pipe (every 10 minutes)
(11) Temperature of condensate (every 10 minutes)
(12) Room temperature (taken twice per test)
(13) Barometric pressure (taken once per test)
(14) Weight of condensate from condenser
(15) Weight of condensate from mixing chamber

The total steam admitted is determined in three parts: first, that which is condensed in the mixing chamber and there separated from the remaining mixture; second, that which is condensed in the condenser; and third, that which is contained in the air saturated with water vapor leaving the condenser at atmospheric pressure and the temperature maintained in the condenser. With an ample supply of cold condensing water and a consequent low exhaust temperature, the actual weight of moisture in the exhaust from the condenser is small. For great precision in measuring the steam supplied, the weight of vapor in the air entering the mixing chamber should be subtracted from the total weight of steam determined as described above. Because of difficulty in determining the exact condition of the compressed air in the air main, and the fact that the error in weight of steam should be small even though the air was initially saturated, the air for the sake of simplicity was assumed to be dry. With the air supply at 75 pounds per square inch absolute and 100 degrees Fahrenheit, the error due to this assumption is approximately 2 per cent of the weight of vapor in the exhaust air at 200 degrees Fahrenheit and a small fraction of one per cent of the total weight of steam used in the process.

21. *Discussion of Observed and Calculated Data.*—The heat content per pound of steam added was determined from the calorimeter data Items 7 and 8, Table 11, and recorded as Item 10; this multiplied by the total condensate, Item 15, of the same table, which is the sum of the condensate from the condenser, the condensate from the separator, and the moisture carried away in the air, gives the total heat energy in the steam added. While it may seem that Item 4 of Table 12, which gives the total steam in terms of equivalent dry steam at the throttle pressure, should be obtained by dividing the total heat energy in the steam by the heat content of dry steam at throttle

pressure, this is not the case. Although in all of these computations the air was supposed to be delivered to the mixer at 100 degrees Fahrenheit, Item 6, Table 11, shows that this was not actually true. A correction was therefore necessary. If the air entered the mixer below 100 degrees Fahrenheit more steam was actually used than if it had entered at the standard temperature; hence the heat required to raise the air used in the test from the temperature entering the mixer to 100 degrees Fahrenheit was subtracted from the total heat of the steam used. Similarly when the air entered the mixer at a temperature above 100 degrees Fahrenheit less steam was actually used than when the air temperature was 100 degrees; hence the correction was added. The total heat thus corrected and divided by the heat content of dry steam at throttle pressure gives Item 4 of Table 12. By means of this correction it is possible in subsequent calculations to disregard the actual temperature of the air entering the mixer. From the previous air tests, air-rates per indicated horse power and brake horse power are known.

The manometer readings, Item 4, Table 11, are very small when the per cent of steam in the mixture is large. When the air consumption was small the mercury in the manometer was replaced by water which clearly showed small differences of pressure. In the table these pressures were converted from the observed heights of the column of water to heights of a column of mercury. From Item 6 of Table 12 it is seen that after all corrections were made the ratio of the steam used to the total mixture often deviated considerably from the predetermined ratio as shown by the last number of the test designation. This deviation, however, caused no difficulty. Items 7 and 8 show that the temperature difference between the throttle and the steam chest is small compared with that found when air alone is used. Even a small quantity of steam, 10 to 15 per cent of the mixture, will make this difference 4 degrees Fahrenheit or less; in the

TABLE 11

OBSERVED VALUES FOR ENGINE TESTS WITH AIR-STEAM MIXTURES

		Venturi Meter					Throttling Calorimeter			
	1	2	3	4	5	6	7	8	9	10
Designation	Date	Temperature of Air Upstream (deg. F.)	Pressure of Air Upstream (lb. per sq. in. Gage)	Venturi Meter Manometer (in. Mercury)	Barometer (in. Mercury)	Temperature of Air Entering Mixer (deg. F.)	Pressure at Calorimeter (lb. per sq. in. Gage)	Temperature at Calorimeter (deg. F.)	Quality of Steam	Heat Content of Steam B. t. u.
A-S-100-10	5-5-13	128.6	101.3	1.42	29.31	106.7	121.4	250.4	0.972	1168
A-S-100-20	5-5-13	137.4	101.0	0.91	29.30	113.4	122.1	251.0	0.972	1168
A-S-100-30	5-5-13	136.0	102.9	0.64	29.30	113.0	120.1	249.9	0.971	1167
A-S-100-50	5-6-13	101.5	101.2	0.35	29.45	87.2	120.4	249.3	0.972	1167
A-S-100-75	5-6-13	102.3	100.5	0.12	29.45	89.2	120.6	249.9	0.972	1167
A-S-100-80	5-6-13	97.0	100.0	0.03	29.45	91.2	121.9	249.9	0.971	1167
S-100	5-6-13	....	....	....	29.45	....	100.0	255.6	0.980	1171
A-S-80-10	4-25-13	106.9	81.9	1.32	29.35	89.4	110.5	249.3	0.977	1170
A-S-80-20	4-25-13	96.1	81.1	0.90	29.35	80.2	110.0	249.7	0.977	1170
A-S-80-22.3	4-21-13	104.8	82.4	0.83	29.41	83.1	114.2	248.2	0.973	1167
A-S-80-30	4-28-13	98.0	80.7	0.60	29.21	82.3	110.6	249.5	0.975	1170
A-S-80-50	4-28-13	99.5	80.7	0.32	29.21	84.3	109.9	250.1	0.976	1170
A-S-80-75	4-28-13	93.4	80.1	0.13	29.21	84.3	110.2	249.3	0.975	1169
A-S-80-80	4-30-13	91.0	77.1	0.04	29.37	87.5	110.1	251.1	0.975	1169
S-80	4-25-13	....	....	....	29.35	....	80.0	248.6	0.980	1167
A-S-60-10	4-15-13	118.7	61.6	0.99	29.20	85.1	113.9	254.6	0.977	1170
A-S-60-20	4-15-13	101.1	62.7	0.62	29.20	76.9	110.0	256.4	0.979	1171
A-S-60-30	4-15-13	106.4	61.8	0.46	29.20	82.2	111.3	255.5	0.978	1170
A-S-60-50	4-16-13	106.1	62.1	0.26	29.14	84.8	114.4	248.1	0.972	1166
A-S-60-75	4-16-13	106.3	61.0	0.11	29.14	87.2	109.9	247.3	0.972	1166
A-S-60-80	4-21-13	83.4	60.4	0.03	29.46	74.8	106.3	243.2	0.971	1165
S-60	4-21-13	....	....	....	29.46	....	60.0	231.4	0.975	1166

TABLE 11 (CONTINUED)

OBSERVED VALUES FOR ENGINE TESTS WITH AIR STEAM MIXTURES

Designation	Condensate					i. h. p. Data					
	11	12	13	14	15	16	17	18	19	20	21
	Temperature of Condensate (deg. F.)	Condensate Weighed (lb. per hr.)	Moisture Carried Away in Saturated Exhaust Air of Mixture (lb. per hr.)	Condensate in Separator Due to Heating Air (lb. per hr.)	Total Wet Steam (lb. per hr.)	m. e. p. Head End of Cylinder	m. e. p. Crank End of Cylinder	i. h. p. Head End of Cylinder	i. h. p. Crank End of Cylinder	Revolutions (per min.)	Net Weight on Scales
A–S–100–10	71.4	96.6	9.3	6.9	112.8	46.0	46.5	10.97	10.51	104.2	191
A–S–100–20	82.6	162.5	10.4	5.7	178.6	45.4	46.7	10.85	10.57	104.4	191
A–S–100–30	83.6	207.7	8.2	6.6	222.5	45.7	47.0	10.91	10.65	104.4	191
A–S–100–50	85.1	290.0	7.2	6.2	303.4	45.4	48.0	10.82	10.84	104.1	196
A–S–100–75	88.9	423.6	4.8	4.6	433.0	47.3	49.6	11.26	11.18	104.0	199
A–S–100–80	90.4	500.8	2.7	1.8	505.3	47.8	48.7	11.39	11.00	104.1	199
S–100	96.0	630.4	0.0	0.0	630.4	47.6	51.4	11.33	11.59	103.9	206
A–S–80–10	80.4	76.1	8.1	5.7	89.9	36.7	38.6	8.74	8.71	104.0	151
A–S–80–20	82.0	111.2	6.0	5.4	122.6	35.4	37.6	8.42	8.48	103.9	151
A–S–80–22.3	111.6	105.5	22.4	5.8	133.7	36.8	37.2	8.78	8.41	104.2	154
A–S–80–30	92.7	170.3	6.9	4.7	181.9	35.5	38.2	8.45	8.61	103.9	151
A–S–80–50	101.3	248.7	4.3	4.9	257.9	35.6	38.1	8.47	8.59	103.9	151
A–S–80–75	101.2	317.3	2.7	4.2	324.2	36.4	36.9	8.68	8.33	104.1	151
A–S–80–80	92.0	392.0	2.2	3.5	397.7	35.7	37.7	8.50	8.51	104.0	151
S–80	123.1	532.0	0.0	0.0	532.0	38.1	40.1	9.08	9.06	104.1	161
A–S–60–10	65.7	567.0	4.7	4.0	65.4	26.2	27.4	6.25	6.19	104.2	106
A–S–60–20	78.0	975.0	5.9	4.0	107.4	25.8	26.7	6.16	6.04	104.2	106
A–S–60–30	84.0	130.3	5.9	3.8	140.0	25.7	26.3	6.14	5.96	104.4	106
A–S–60–50	107.6	178.0	9.7	3.6	191.3	25.6	26.6	6.11	6.01	104.2	106
A–S–60–75	117.9	242.5	7.9	2.9	253.3	26.1	27.0	6.22	6.10	104.1	106
A–S–60–80	91.0	311.5	2.0	2.6	316.1	25.2	27.5	6.00	6.21	104.0	106
S–60	98.7	397.5	0.0	0.0	397.5	24.6	27.0	5.86	6.10	104.1	106

TABLE 12

OBSERVED VALUES AND COMPUTED RESULTS FOR ENGINE TESTS WITH AIR-STEAM MIXTURES

Designation	1	2	3	4	5	6	7	8	9	10	11	12	13
	Pressure at Throttle lb. per sq. in. Gage)	Total Air		Total Steam, Terms of Dry Steam (Pres. in Item 1) lb. per hr.	Total Mixture lb. per hr. (Item 3 + Item 4)	Dry Steam in % of Mixture	Temperature (deg. F.)			i. h. p.	b. h. p.	Mech. Efficiency	cu. ft. Free Air per min. per i. h. p. (62° F., 14.7 lb. per sq. in. Abs.)
		cu. ft. Free Air per min. (60° F., 14.7 lb. per sq. in. Abs.)	lb. per hr.				Mixture at Throttle	In Steam Chest	At Exhaust				
A–S–100–10	100	144	654	112	766	14.6	234	231	128	21.48	19.90	92.6	6.65
A–S–100–20	100	114	522	177	701	25.2	268	266	158	21.42	20.00	93.2	5.32
A–S–100–30	100	98	444	220	664	33.1	284	282	171	21.56	20.00	92.9	4.51
A–S–100–50	100	73	334	296	630	47.0	302	300	185	21.66	20.44	94.5	3.37
A–S–100–75	100	43	197	423	620	68.2	321	319	201	22.44	20.70	92.2	1.92
A–S–100–80	100	23	105	494	599	82.5	330	329	207	22.39	20.74	92.8	1.03
S–100	100	0	0	622		100.0	338	338	212	22.92	21.40	93.4	0.00
A–S–80–10	80	129	586	87	673	12.9	222	218	125	17.45	15.70	90.1	7.37
A–S–80–20	80	107	491	119	611	19.5	244	241	147	16.90	15.68	92.9	6.34
A–S–80–22.3	80	103	472	130	602	21.6	249	245	151	17.19	16.02	93.2	6.00
A–S–80–30	80	38	400	178	578	30.8	267	265	169	17.06	15.68	92.0	5.13
A–S–80–50	80	64	290	254	544	46.7	288	286	186	17.06	15.68	92.0	3.73
A–S–80–75	80	39	180	318	498	63.9	303	302	197	17.01	15.71	92.5	2.33
A–S–80–80	80	24	107	391	498	78.5	313	312	205	17.01	15.70	92.5	1.38
S–80	80	0	0	525	...	100.0	326	225	212	18.14	16.75	92.5	0.00
A–S–60–10	60	102	448	63	511	12.3	201	198	121	12.44	11.05	89.0	7.50
A–S–60–20	60	80	367	105	472	22.2	235	233	151	12.20	11.05	90.7	6.38
A–S–60–30	60	67	307	138	445	31.0	251	248	164	12.10	11.08	91.6	5.57
A–S–60–50	60	52	236	189	425	44.5	271	269	182	12.12	11.05	91.3	4.26
A–S–60–75	60	32	146	250	396	63.1	287	285	196	12.32	11.03	89.5	2.61
A–S–60–80	60	19	82	311	393	79.1	298	297	205	12.21	11.02	90.4	1.46
S–60	60	0	0	390	...	100.0	309	307	212	11.96	11.02	92.4	0.00

14	15	16	17	18	19	20	21	22	23	24	25	26	27	28
Air lb. per hr. per i. h. p.	Steam lb. per hr. per i. h. p.	Mixture lb. per hr. per i. h. p.	cu. ft. Free Air per min. per b. h. p. (62° F., 14.7 lb. per sq. in. Abs.)	Air lb. per hr. per b. h. p.	Steam lb. per hr. per b. h. p.	Mixture lb. per hr. per b. h. p.	Barometer (in. Mercury)	Room Temperature (deg. F.)	B. t. u. Added per hr. by Steam = Heat Content of Steam in Item 4— Heat of Liquid at Exhaust Temp.	lb. Air Req. per i. h. p. hr. at 100° F., and Pressure in Item 1	i. h. p. hr. Obtainable from Air (Item 3) 100° F. Pressure in Item 1 (Item 3 ÷ Item 24)	i. h. p. hr. Added by Steam (Item 10 Less Item 25)	B. t. u. Req. per i. h. p. hr. Added by Steam (Item 23 + Item 26)	lb. Dry Steam Used per i. h. p. hr. Added (Item 4 + Item 26)
30.3	5.2	35.5	7.16	32.7	5.6	38.3	29.3	81	122 010	45.3	14.36	7.12	17 120	15.7
24.3	8.3	32.6	5.72	26.1	8.9	35.0	29.3	83	187 930	45.3	11.50	9.92	18 940	17.8
20.6	10.2	30.8	4.87	22.2	11.0	33.2	29.3	86	230 850	45.3	9.81	11.75	19 660	18.7
15.5	13.6	29.1	3.57	16.3	14.5	30.8	29.5	78	306 500	45.3	7.38	14.28	21 410	20.7
8.8	18.8	27.6	2.08	9.5	20.4	29.9	29.5	78	431 500	45.3	4.35	18.09	23 810	23.4
4.7	22.0	26.7	1.11	5.1	23.8	28.9	29.5	80	501 190	45.3	2.32	20.07	24 980	24.6
0.0	27.1	....	0.00	0.0	29.0	....	29.5	81	627 500	....	....	22.92	27 230	27.1
..	5.0	38.6	8.18	37.3	5.5	42.8	29.4	70	95 350	48.0	12.19	5.26	18 100	16.5
..	7.1	36.1	6.85	31.3	7.6	38.9	29.4	69	127 240	48.0	10.19	6.71	19 020	17.7
..	7.6	35.0	6.41	29.5	8.1	37.6	29.4	82	138 440	48.0	9.83	7.36	18 780	17.7
..	10.4	33.8	5.57	25.5	11.3	36.8	29.2	78	186 350	48.0	8.32	8.74	21 310	20.4
17.0	14.9	31.9	4.05	18.5	16.2	34.7	29.2	77	261 440	48.0	6.04	11.02	23 740	23.0
10.6	18.7	29.3	2.50	11.4	20.2	31.6	29.2	80	324 840	48.0	3.76	13.25	24 430	24.0
3.5	22.9	29.2	1.48	6.8	24.9	31.7	29.4	85	395 690	48.0	2.24	14.77	26 750	26.5
0.0	28.9	....	0.00	0.0	31.3	....	29.4	70	527 800	....	....	18.14	28 950	28.9
36.1	5.0	41.1	8.88	40.5	5.7	46.2	29.2	75	69 050	51.1	8.77	3.67	18 790	17.1
30.1	8.6	38.7	7.28	33.2	9.5	42.7	29.2	73	111 150	51.1	7.17	5.03	22 080	20.9
25.5	11.4	36.9	6.10	27.8	12.4	30.2	29.2	75	144 230	51.1	6.02	6.08	23 710	22.7
19.5	15.6	35.1	4.69	21.3	17.1	38.4	29.1	79	194 400	51.1	4.61	7.51	25 770	25.2
11.8	20.3	32.1	2.91	13.3	22.6	35.9	29.1	83	253 810	51.1	2.86	9.46	26 790	26.4
6.7	25.5	32.2	1.65	7.5	28.2	35.7	29.5	76	313 720	51.1	1.61	10.60	29 580	29.4
0.0	32.6	....	....	0.0	35.3	....	29.5	79	390 600	....	....	11.96	32 540	32.6

29	30	31	32	33	34	35	36	37	38	39	40	41	42	43
Engine Water-Rate Pressure in Item 1 25% Cut-off	i. h. p. hr. Obtainable from Steam (Item 4) (Item 4 + Item 29)	i. h. p. hr. Obtainable Using Air and Steam Separately (Item 25 + Item 30)	i. h. p. Gained by Mixing (Item 10 Less Item 31)	Per Cent Gain (Item 32 ÷ Item 31)	lb. Air Req. per b. h. p. hr. at 100° F. and Pressure in Item 1	b. h. p. hr. Obtainable from Air (Item 3) 100° F. Pressure in Item 1 (Item 3 ÷ Item 24)	b. h. p. Added by Steam (Item 11 Less Item 35)	B. t. u. Req. per b. h. p. hr. Added by Steam (Item 23 ÷ Item 36)	lb. Dry Steam Used per b. h. p. hr. Added (Item 4 ÷ Item 36)	Engine Water-Rate Pressure in Item 1 25 % Cut-off	b. h. p. hr. Obtainable from Steam (Item 4) (Item 4 + Item 39)	b. h. p. hr. Obtainable Using Air and Steam Separately (Item 35 + Item 40)	b. h. p. Gained by Mixing (Item 11 Less Item 41)	Per Cent Gain (Item 42 + Item 41)
27.1	4.14	18.50	2.98	16.1	54.4	11.97	7.93	15 390	14.1	29.0	3.87	15.84	4.06	25.6
27.1	6.55	18.05	3.37	18.7	54.4	9.60	10.40	18 050	17.0	29.0	6.11	15.71	4.29	27.2
27.1	8.13	17.94	3.62	20.2	54.4	8.18	11.82	19 520	18.6	29.0	7.60	15.78	4.22	26.7
27.1	10.92	18.30	3.36	18.3	54.4	6.15	14.29	21 430	20.7	29.0	10.20	16.35	4.09	25.0
27.1	15.62	19.97	2.47	12.4	54.4	3.62	17.08	25 250	24.8	29.0	14.60	18.22	2.48	13.6
27.1	18.23	20.55	1.84	9.0	54.4	1.93	18.81	26 620	26.2	29.0	17.05	18.98	1.76	9.3
27.1	....	....	....	....	....	....	21.40	29 300	29.0	29.0	....	....	....	....
28.9	3.01	15.20	2.25	14.8	60.8	9.64	6.06	15 700	14.3	31.3	2.78	12.42	3.28	26.4
28.9	4.12	14.31	2.59	18.1	60.8	8.09	7.59	16 780	15.7	31.3	3.81	11.90	3.78	31.8
28.9	4.50	14.33	2.86	19.9	60.8	7.78	8.24	16 800	15.8	31.3	4.16	11.94	4.08	34.2
28.9	6.16	14.48	2.58	17.8	60.8	6.59	9.09	20 500	19.6	31.3	5.70	12.29	3.39	27.6
28.9	8.80	14.84	2.22	14.9	60.8	4.79	10.89	24 000	23.3	31.3	8.12	12.91	2.77	21.4
28.9	11.00	14.76	2.25	15.2	60.8	2.97	12.74	25 470	24.9	31.3	10.15	13.12	2.59	19.7
28.9	13.53	15.77	1.24	7.9	60.8	1.76	13.94	28 370	28.0	31.3	12.50	14.26	1.44	10.1
28.9	....	....	....	....	....	....	16.75	31 500	31.3	31.3	....	....	....	....
32.6	1.93	10.70	1.74	16.3	73.1	6.14	4.91	14 030	12.8	35.3	1.78	7.92	3.13	39.3
32.6	3.22	10.39	1.81	17.4	73.1	5.02	6.03	18 460	17.4	35.3	2.98	8.00	3.05	38.1
32.6	4.24	10.26	1.84	17.9	73.1	4.21	6.87	21 000	20.1	35.3	3.91	8.12	2.96	36.4
32.6	5.81	10.42	1.70	16.3	73.1	3.23	7.82	24 850	24.2	35.3	5.35	8.58	2.47	28.8
32.6	7.38	10.54	1.78	16.9	73.1	2.00	9.03	28 100	27.7	35.3	7.08	9.08	1.95	21.5
32.6	9.56	11.17	1.04	9.3	73.1	1.13	9.89	31 720	31.4	35.3	8.81	9.94	1.08	10.9
32.6	....	....	....	....	....	....	11.03	35 400	35.3	35.3	....	....	....	....

Table 12 (continued)

Designation
A-S-100-10
A-S-100-20
A-S-100-30
A-S-100-50
A-S-100-75
A-S-100-80
A-S-100
A-S-80-10
A-S-80-20
A-S-80-22.3
A-S-80-30
A-S-80-50
A-S-80-75
A-S-80-80
A-S-80
A-S-50-10
A-S-50-20
A-S-50-30
A-S-50-50
A-S-50-75
A-S-50-80
A-S-50

air tests at high temperatures this difference was often more than 40 degrees.

Item 6, the per cent of steam in the mixture, plotted against Item 7, the temperature at the throttle, for each total pressure gives the curves shown in Fig. 17. These curves suggest a method for regulating the proportions of the mixture without weighing the condensate. With a given cut-off and brake load, and for a specified total pressure, the valves in the air and steam lines can be adjusted to give the temperature at the throttle which results from the desired mixture.

In these air-steam tests the exhaust temperature, Item 9, is reliable and probably in no case is it more than 1 or 2 degrees in error. In all these tests it was considerably above the temperature of the atmosphere and surrounding objects; when steam alone was used, exhausting at atmospheric pressure, the thermometer read 212 degrees Fahrenheit. These exhaust temperatures show that the steam contained in an air-steam mixture is used more efficiently than it would be in a non-condensing steam plant; and, in the case of mixtures having small percentages of steam, it is used with efficiencies comparable to those obtained with condensing engines. In test A–S–100–10 the exhaust temperature was 128 degrees Fahrenheit, which is the temperature that would result in a condensing plant maintaining a 25 inch vacuum. The exhaust temperature may be regarded as a measure of the condenser effect produced from the use of mixtures of air and steam, resulting from the fact that the total exhaust pressure is the sum of the partial pressures of the steam and air.

When even a small quantity of steam was admitted to the cylinder with the air the difficulties with lubrication and the excessive internal friction were overcome, as shown by the indicated and brake horse power, Items 10 and 11. The difference between these items gives a friction horse power of about 1.5 for 100 pounds, 1.4 for 80 pounds, and 1.2 for 60 pounds. In the air tests with serial numbers 22-31 and 48-51, in which the air was 400 degrees at the throttle, about the same indicated horse power was developed as in the air-steam tests at 100 pounds gage. In the air tests, however, the friction horse power was 1.7, showing that even with 400 degrees at the throttle the internal friction was not quite reduced to its minimum value, which, when the engine was developing 21 indicated horse power with 100 pounds pressure at the throttle, may be taken as 1.5 horse power.

Item 12 shows that when the load is nearly constant the mechanical efficiency does not vary with the per cent of steam in the mixture. When the initial pressure is decreased and the cut-off kept constant there is a slight decrease in the efficiency due to decrease of the load, as follows:

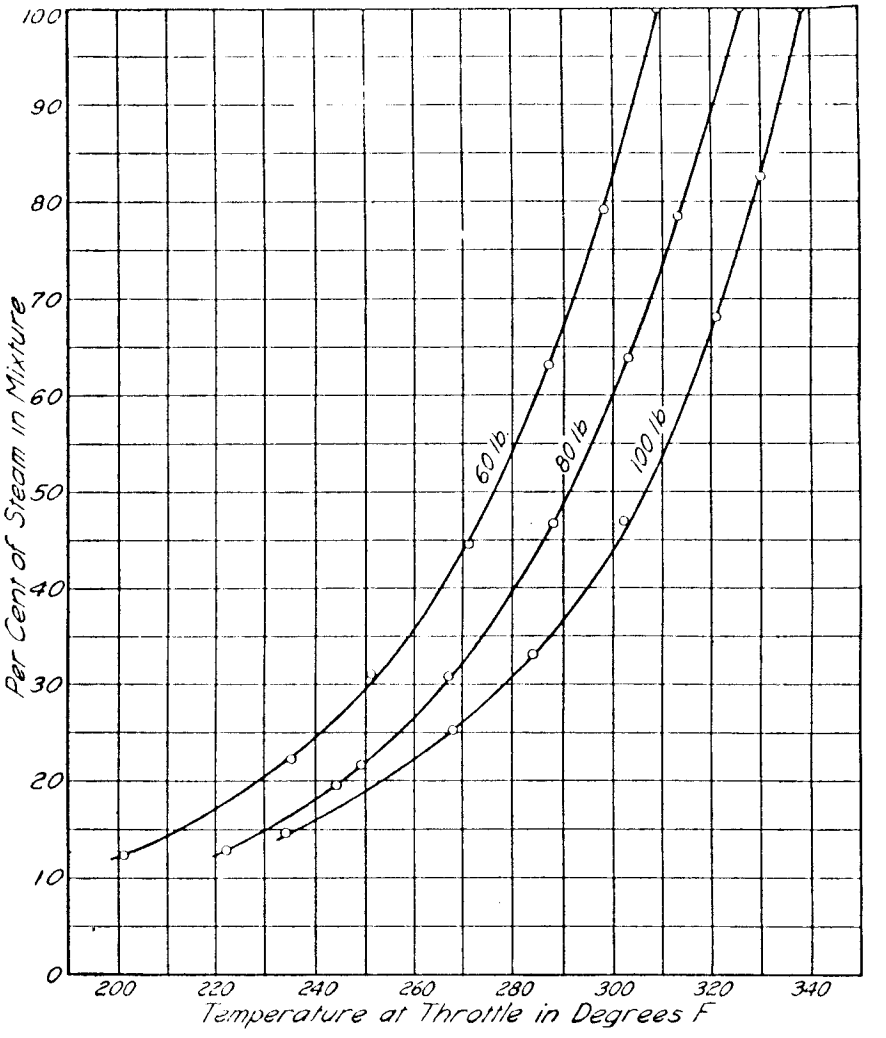

FIG. 17. RELATION BETWEEN TEMPERATURES AND PRESSURES AT THROTTLE AND PERCENTAGES OF STEAM IN AIR STEAM MIXTURE

Pressure at Trottle (gage)	Average i.h.p.	Average Mechanical Efficiency
100	21.98	93.1
80	17.23	92.2
60	12.19	90.7

The air, steam, and mixture rate per indicated horse power given in Items 14, 15, and 16 are plotted to per cent of mixture in Figures 18, 19, 20. In each figure, curve (2) represents the air weight, curve (3) the steam weight, and curve (1) the weight of the mixture.

Curve (1) is therefore the summation curve of curves (2) and (3). Figure 21 is the curve (1) of Figures 18, 19, and 20 drawn to a larger scale. It is unnecessary to plot the corresponding curves for brake horse power since the air and steam consumption per brake horse power per hour may be obtained from these curves by dividing particular values for a given pressure by the average mechanical efficiency for that pressure as shown above.

22. *Economic Results.*—The presentation of the economic effects resulting from mixing air and steam begins with Item 23, which gives the total heat above the exhaust temperature of the equivalent steam supplied during the test. This is the total heat chargeable to the engine, since the condensed exhaust steam is assumed to be used as boiler feed water. In Item 25 the indicated horse power per hour obtainable from the air in the mixture if used alone at 100 degrees Fahrenheit is found by dividing Item 3 by Item 24, which is the air-rate per indicated horse power at the given throttle pressure and 100 degrees Fahrenheit determined from air tests. Subtracting Item 25 from Item 10 gives the indicated horse power which may be assumed to

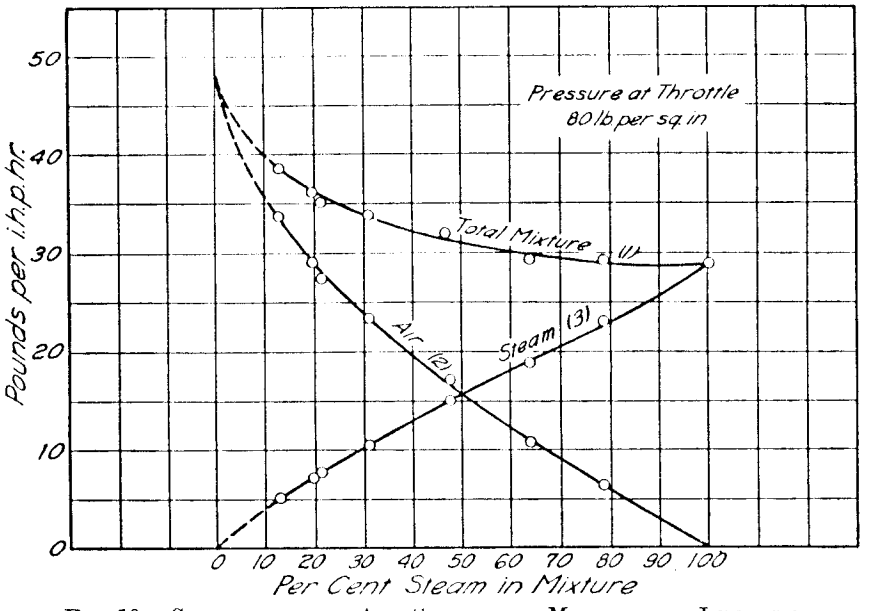

FIG. 19. CONSUMPTION OF AIR, STEAM, AND MIXTURE PER INDICATED HORSE POWER PER HOUR FOR DIFFERENT PERCENTAGES OF STEAM AND MIXTURE; THROTTLE PRESSURE 80 LB. PER SQ. IN.

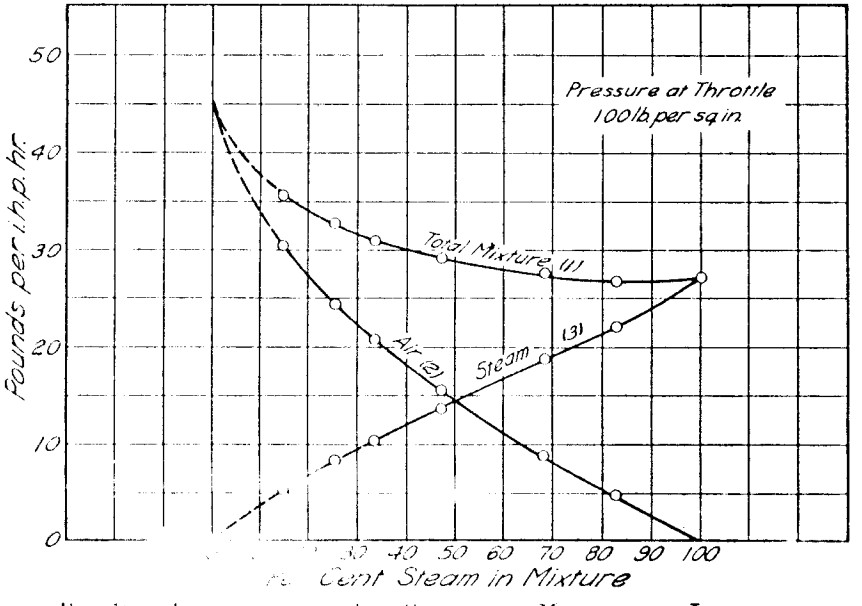

FIG. 18. CONSUMPTION OF AIR, STEAM, AND MIXTURE PER INDICATED HORSE POWER PER HOUR FOR DIFFERENT PERCENTAGES OF STEAM AND MIXTURE; THROTTLE PRESSURE 100 LB. PER SQ. IN.

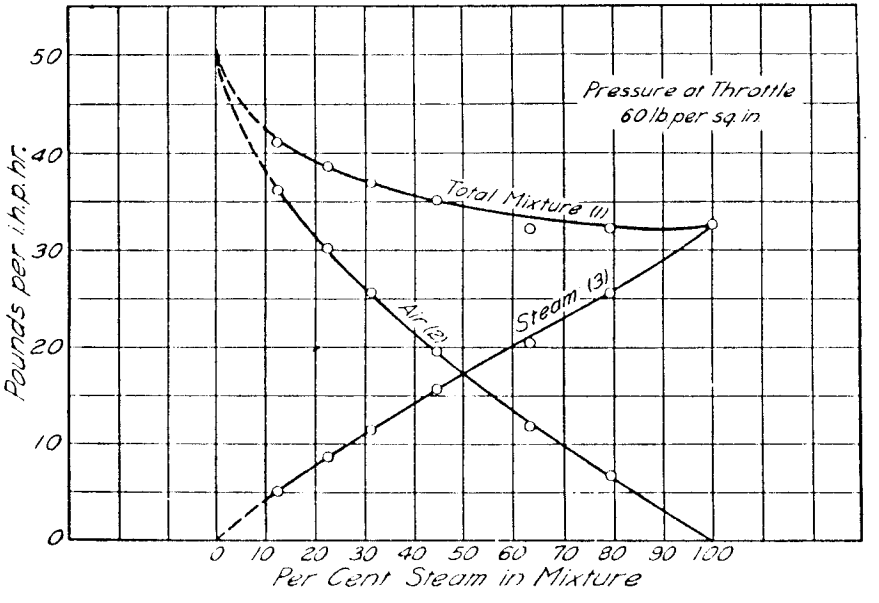

FIG. 20. CONSUMPTION OF AIR, STEAM, AND MIXTURE PER INDICATED HORSE POWER PER HOUR FOR DIFFERENT PERCENTAGES OF STEAM AND MIXTURE; THROTTLE PRESSURE 60 LB. PER SQ. IN.

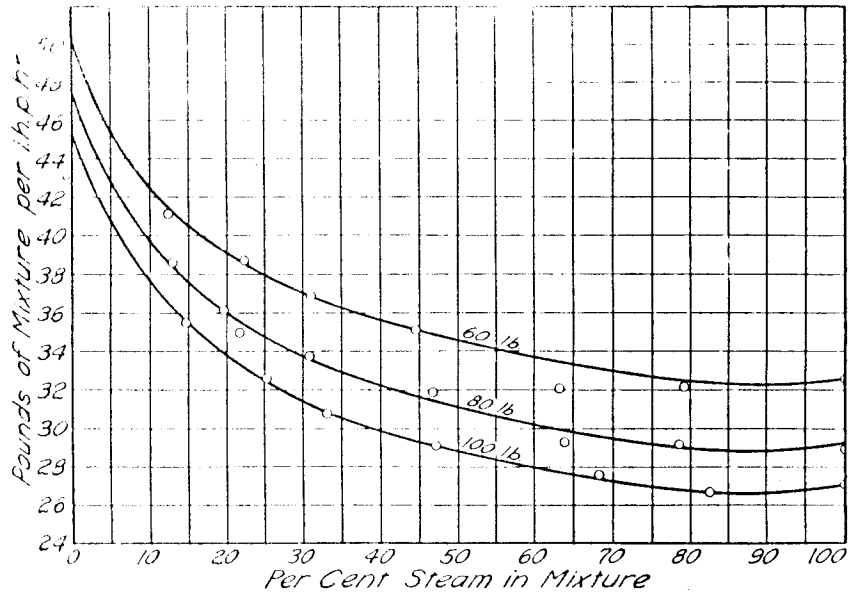

FIG. 21. CONSUMPTION OF AIR, STEAM, AND MIXTURE PER INDICATED HORSE POWER PER HOUR FOR DIFFERENT PERCENTAGES OF STEAM AND MIXTURE

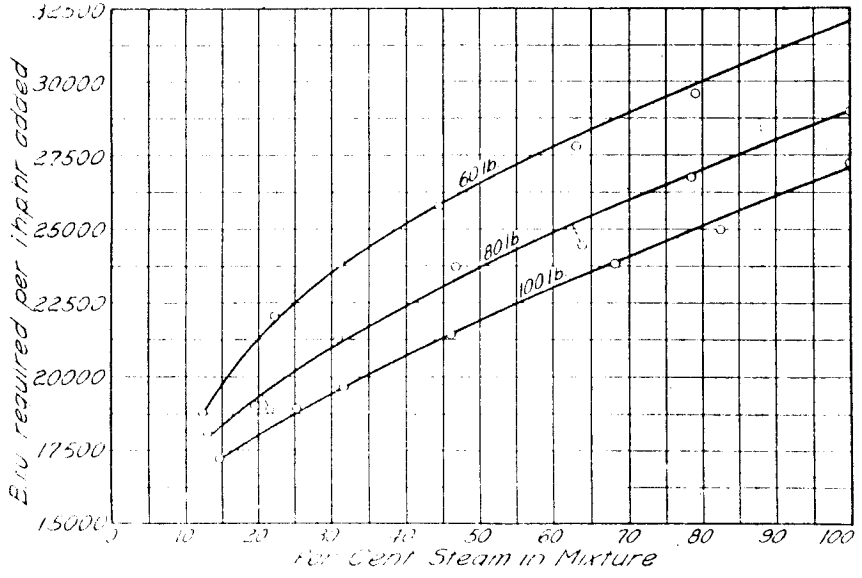

FIG. 22. HEAT REQUIRED PER ADDITIONAL INDICATED HORSE POWER PER HOUR FOR DIFFERENT THROTTLE PRESSURES AND DIFFERENT PERCENTAGES OF STEAM IN MIXTURE

be chargeable to the steam in the mixture. Items 27 and 28 give the cost of the added horse power in terms of British thermal units and pounds of dry steam. These two items increase steadily as the per cent of steam is increased. With 15 per cent of steam in the mixture the number of British thermal units required per indicated horse power per hour added is slightly more than one-half the number required per indicated horse power per hour when steam alone is used. Again, comparing the tests with 15 per cent steam with air tests at the same pressure, it is seen that the number of British thermal units required per indicated horse power per hour added in the case of air-steam mixtures is about twice as great as the number required per indicated horse power per hour gained by reheating air alone. Figure 22 shows the characteristics here discussed.

Item 29 shows the best obtainable steam consumption in pounds per indicated horse power per hour, with cut-off at 25 per cent of the stroke and an initial pressure equal to the total initial pressure of the mixture in a particular test, as determined in a series of tests using steam alone. If the engine had been operated at full load under the conditions specified in Item 29 until the weight of steam given in Item 4 was used, the number of indicated horse power hours shown in Item 30 would have been obtained. By a similar process the number of indicated horse power hours which could have been secured with the actual weight of air used in a particular test is given in Item 25. Therefore the sum of Items 25 and 30, Item 31, represents the power which could have been realized in the engine tested if the air and steam in the mixture had been used separately. The actual power secured from the mixture of air and steam in a particular test is shown under Item 10. Hence the difference between Items 10 and 31, Item 32, is the gain in power, in indicated horse power hours, which results when the engine is operated with the given mixture of the air and steam. Through this method of reasoning it is evident that the mixture of the air and steam affords a thermodynamic advantage over the use of the separate ingredients amounting to as much as 20.2 per cent, based on indicated horse power, (Test A–S–100–30), or 39.6 per cent based on brake horse power, (Test A–S–60–10).

By a similar process of calculation the economic results of the tests, based on brake horse power, are calculated and presented in Table 12 under Items 34 to 43 inclusive.

The astonishing results of these tests of air-steam mixtures must not be construed as an argument for the use of such a medium in the generation of power. If the air is compressed by a steam driven compressor the total steam consumption per horse power produced by an engine using a mixture of air and steam would probably be

greater than through the use of steam alone, when the steam used in the compressor is included. When, however, a supply of compressed air is available and is used for generating power, and steam is also available, the mixture of the two affords a distinct advantage over the use of air alone or of steam alone if the cost of compressing the air is disregarded.

VI. General Summary and Conclusions

23. *Reheaters and Reheating.*—

(a) Commercial types of small external combustion reheaters will under favorable conditions and when operated at their best capacity supply 60 per cent of the heat in the fuel to the compressed air.

(b) The efficiency is dependent upon the design of the heating surface and its relation to the furnace or fire-pot. The maximum efficiency in improperly designed apparatus may be very low. Heating surface in direct contact with the fire-pot is three times as effective as such surface not in direct contact with the fire or very close to it.

(c) Internal combustion reheaters may attain efficiencies of 80 per cent or more when operating at their best capacity with reheated air temperatures of approximately 400 degrees Fahrenheit.

(d) The efficiencies of all types of reheaters decrease rapidly when the reheater is operated at low capacities. The size of the reheater should therefore be carefully selected with reference to the average capacity demanded by the service.

(e) The results of this investigation indicate that a chimney draft of from 0.15 to 0.25 inches of water is essential for the best results.

(f) One thousand pounds of air per hour or 219 cubic feet of free air per minute can be heated through a range of 100 degrees Fahrenheit with 3¼ to 3½ pounds of fuel whose calorific value is 13 000 British thermal units per pound.

(g) To avoid excessive losses of heat the reheater must be installed as close as possible to the motor it serves.

24. *Expansive Use of Reheated Air.*—

(a) With the particular engine employed in this investigation operating with a cut-off at one-quarter stroke the air consumption per indicated horse power per hour decreases as follows, as the air temperature rises from 100 degrees to 400 degrees Fahrenheit:

With an initial gage pressure of 100 pounds per square inch the air consumption varies from approximately 45 pounds to 33 pounds, or from 10 cubic feet to 7.3 cubic feet of free air per minute.

With an initial gage pressure of 80 pounds per square inch the air consumption varies from approximately 48 pounds to 35.23 pounds or 10.5 cubic feet to 7.73 cubic feet of free air per minute.

With an initial gage pressure of 60 pounds per square inch the air consumption varies from approximately 51.1 pounds to 38.3 pounds, or 11.1 to 8.45 cubic feet of free air per minute.

(b) Similarly the air consumption per brake horse power per hour with other conditions as given above varies as follows:

at 100 pounds gage pressure, from 58.3 pounds to 36.2 pounds per hour, or from 12.77 cubic feet to 7.93 cubic feet of free air per minute;

at 80 pounds gage pressure, from 63.35 pounds to 39.45 pounds per hour, or from 13.9 cubic feet to 8.65 cubic feet of free air per minute;

at 60 pounds gage pressure, from 76.43 pounds to 45.84 pounds per hour, or 16.7 cubic feet to 10 cubic feet of free air per minute.

(c) Air at 400 degrees Fahrenheit will yield the same brake power with two-thirds the compressor capacity and steam consumption required when the air is used at 100 degrees Fahrenheit.

(d) When air is reheated from 100 degrees to 400 degrees Fahrenheit an indicated horse power per hour is gained by the expenditure of approximately 8700 to 9300 British thermal units, or 1.15 pounds of fuel, whose heating value is 13 000 British thermal units, burned in the reheater with 60 per cent efficiency. The heat required for the first 100 degree rise in temperature is particularly effective. In this range an indicated horse power was gained by the addition of about 7700 British thermal units or less than one pound of fuel.

(e) Reheating the air from 100 degrees to 400 degrees Fahrenheit permits the gain of a brake horse power per hour with an expenditure of heat as low as 8500 British thermal units. In Test 3-100-200-15 a brake horse power was gained with an expenditure of only 7035 British thermal units or about 0.9 pounds of fuel. Test 44-60-200-25 shows an even more astonishing result for a rise of temperature of 100 degrees only.

(f) For the initial pressures employed in these tests the highest efficiencies were obtained when the cut-off was at about

one quarter of the stroke. Complete expansion does not show equally good results.

(g) The use of reheated air leads to an improvement in lubrication, an astonishing increase in mechanical efficiency, and the prevention of ice formation in motors using air expansively. In practice the chief reason for the use of reheaters is the prevention of interference with the operation of the motor through the accumulation of ice in the exhaust pipe.

25. *Use of Steam for Reheating Air.*—

(a) Steam is a convenient medium for reheating air, and a mixture of air and steam containing as little as 10 per cent of steam will entirely prevent the operating difficulties experienced when cold air is used in expansion motors through defective lubrication and the formation of ice in the exhaust pipe.

(b) A greater amount of work is obtained through the use of a mixture of air and steam than could be realized by the separate use of the same weights of each of the two substances. When the steam constitutes 15 to 20 per cent of the mixture there is a gain of from 16 to 20 per cent in the indicated horse power, and of from 25 to 35 per cent in the brake horse power of the engine. An increase in the weight of steam in the mixture reduces the gain.

(c) A much greater expenditure of heat is required per horse power gained when steam is used than is necessary with other methods of reheating.

"Compressed Air Motors," Carl Snyder, *Harper's Weekly,* December 5, 1896:

For the last three months, on the 125th Street surface line in New York City, two new cars have been in operation that offer at novelty in the way of street traction. They run without visible means of propulsion; there is neither overhead trolley cable nor conduit; they are very near to noiseless, they run smoothly, and start and stop without jerk or jar. These cars alternate with those propelled by cable, and, save in the smoothness of their operation, there is outwardly nothing to announce that they are unique. They have thus far covered 12,000 miles and carried some 75,000 passengers, and this not so much without accident as without incident. They are a long stride toward the dream of the street-railway manager—an independent self-contained motor—and that they are successful means much. If the claims of economy are substantial it is an allowable surmise that this is the traction force of the future in large cities if not in small.

"...if a given quantity of compressed air costs a dollar to generate, the further expenditure of ten cents in reheating will double its power to do work."

The new motors represent the latest phase of development in a very wonderful new power which has been rapidly coming into use in the last ten or fifteen years. That is compressed air. The remarkable powers of this force have been largely lost sight of in our modern flood of marvels. If, a hundred years ago, Franklin had made an announcement of a way to bottle up the ambient atmosphere in steel flasks or flasks of any sort, and to tuck these in obscure corners, and use the power resident in them for many and divers conveniences, there is hardly room for doubt that his discovery would have been regarded in that day every whit as extraordinary as his feat of drawing lightning from the clouds. And there are those of competent judgment who believe that if a tithe of the inventive and mechanical genius that has been employed in the development of electricity had been diverted to the perfection of appliances for the use of compressed air, the present relation of these two novel aids of industry would be very different, if not reversed. But electricity was mantled with the glamour of the unknown; for a long time there seemed no limit to its capabilities; it was being applied everywhere and in the most surprising variety of ways; and in the furor it created, compressed air found little favor beside its more its brilliant rival.

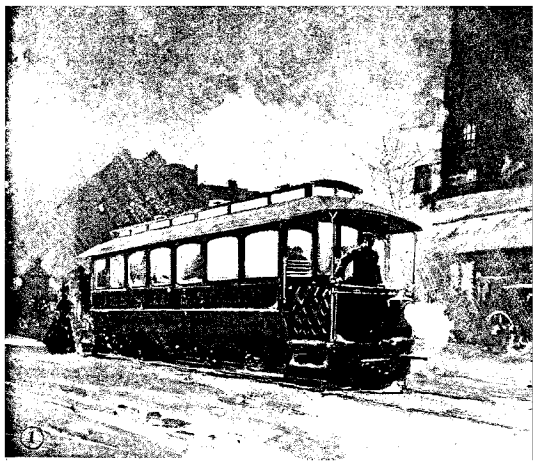

Fig. 1—Car using the Hardie Compessed Air Motor on 125th Street, New York City.

Of recent years, however, the limitations of electricity have become more clearly defined, and in the mean while compressed air has gained a hearing. There are few, perhaps. outside of those who follow the technical journals, who are aware of the immense progress that has of late been made in the employment of this new power, and the astonishing multiplicity of uses to which it is now put.

The Westinghouse air brake, and the compressed-air drill that is to be seen almost any day making excavations for buildings and the like, have long ceased to be regarded as novelties. But it is different when we learn that this same force is now used to start cars, and even to run them, and that this same air-drill, working in the mine, has

literally added hundreds of millions to the available mineral wealth of the world. The enormous increase in the production of gold, as well as of silver, coal and iron, copper, and other metals, has in no small part been due to this convenient tool. Further, it has been the principal agent in the construction of Chicago's great drainage canal, in many respects one of the most wonderful enterprises of the day.

Then again, we are all of us aware of the use of compressed air in the Zalinski dynamite gun, which is able to hurl a charge of explosive a mile and a half, sufficient to convert a whole regiment into unrecognizable pulp. It is another matter to learn of this same force cleaning carpets, dusting cushions, and painting cars and barns. And as we go a little further, we find this Protean force operating block signals on our railroads, and steering ships, running clocks, and furnishing cold air for refrigerators, loading guns and handling projectiles on our men-of-war, propelling sewing-machines, doing all sorts of hoisting-work, driving lathes and printing-presses, copying letters, and running summer fans. In Australia it is shearing sheep; in Kansas City beeves are slaughtered and the meat dressed mainly with compressed air. It is an excellent pump, especially for deep wells, and in particular for chemicals. With the same power you may dump a whole train of coal or dirt cars by the pressure of your thumb. It is carving beautiful statuary, and employed in all sorts of stone-work; it makes a good dredge; it raises and lowers railroad gates; it is a valuable agent in the sugar-refinery and in the making of asphalt and rubber, and still again in the delicate manufacture of fine silk. In the coal-mines it is running locomotives, bringing oxygen and life to the exhausted operative, and banishing the fear of deadly explosions. In England a hundred and fifty miles of pneumatic tubing facilitates the rapid transfer of mails, and the same system is in use in Philadelphia, and just recently between New York and Brooklyn. In England, again, employed in the dry docks, compressed air is made to lift huge vessels out of the water as if they were toys, and in the same way it will shortly be introduced on the Erie Canal in working the new quick-action high-lift locks. It is a good fire-extinguisher, and an excellent hoist for grain. It is used by the physician and surgeon in many delicate operations. In the railroad shops of Jersey City and many other points it is everywhere running machinery, lifting huge loads, riveting bolts, driving hammers, sand-papering cars. In Paris, under the famous Popp system, where it has had its widest application, it is employed in almost every conceivable variety of work.

Now that this new power has, so to speak, found its feet, it would be difficult to set limits upon the vast variety of uses to which, in the future, it will be put. It forms the basis of the gigantic project of a great ship-canal from the Great Lakes to the sea, described in these pages a year or more ago, which will be equipped with locks capable of raising an ocean liner to the height of Niagara with the same ease as a clumsy canal barge is now lifted the height of a bean-pole, and with much more rapidity. The bicycle tire, the pneumatic cushion and pillow and horse collar, are but a beginning of its use in this direction, and it seems now the most convenient power for the automobile cart. And when it is generated in large central stations, and distributed over the city in the same way that gas and water and electricity are now distributed, you may expect, madam, that it will clean house for you—beat your carpets and clean your walls--and take a general hand in your household affairs. It will pick you up and set you down from floor to floor. It will be waiting for you at your door, and whisk you to the shopping districts. It may treadle the sewing-machine, agitate the dish-washer and smash your costliest china with all the dexterity and *sang-froid* of your most accomplished handmaiden. All this and much more it may do, if we are to believe some of those who are given to peering into the to-morrow of things. These tell us that even now compressed air is an equal-footed rival of electricity, and that while

the latter has been thrusting the steam-engine from the field, compressed air has entered the lists against this latest knight of industry.

In view of all these things, its multiplied uses in the present, and the large possibilities of its future, the wonder of it all now seems that compressed air should not have come in long ago. It is not new, any more than electricity is new. In a sketch of its history in an interesting little periodical, Compressed Air, published in New York City, the earliest notable use of this power is set down to the construction of the Mount Cenis tunnel. It was first employed on a large scale in this country in the building of the Hoosac tunnel, along about 1866, and from this time its advancement has been steady and sure. But it has been, apparently, the Cinderella of the mechanical arts. It made its appearance in an age given over to a sort of electrical mania, and there seemed no godmother genius like Tesla or Edison to bring the golden slipper to its neglected hearth.

Three notable steps of advance wrought a striking change in this order of affairs. When compressed air was first tried, it was found that the loss of power was enormous. It was difficult to store, for the air leaked rapidly away; it was expensive to generate, and there were thermo-dynamic difficulties in its use without number. When a thousand cubic feet of air is jammed into the space of one, a large amount of heat is developed, and in order to store and use the air this heat must in some way be drawn off. Similarly, air at high pressure, when released, cools rapidly. The result, if there be a sufficient moisture, is freezing and clogging. For a long time it was thought these difficulties were largely insuperable.

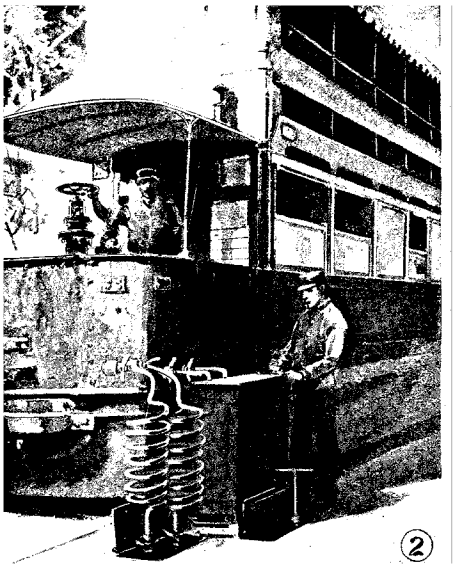

Fig. 2.—The Mékarski Compressed Air System used in Paris—recharging *en route.*

Now, however, these very difficulties are turned to a profit—to such excellent profit, indeed, as to afford an apparent paradox. It seems idle to assert that it is possible to get as much power out of a machine as you put into it—this means a frictionless and wasteless mechanism. And yet a very near approach to just this condition seems to have been made in the case of compressed air. This is due to the development of the reheating process. Lest the reader be not familiar with the technique of the subject, it may not be idle to explain its broader features. In the process of compression the air is sucked into a piston, and then rammed into a reservoir surrounded by a water jacket, the latter drawing off the heat generated in the compression. The machine which does this work is a beautiful affair of what is known as the four-stage type. That is to say, the air is first driven up to about eighty pounds pressure and cooled; then turned into a second cylinder, where it is compressed still further, then cooled again; and so on up to the desired point. Thus even at two or three thousand pounds pressure to the square inch the air within the reservoir remains at somewhere near the temperature of the outside atmosphere. But if the air be used in this condition, not only will a large share of the power employed in compression be lost, but it will, as already noted, have a tendency to freeze everything within reach. If, however, as it is released, it is passed through a heater or is shot through superheated hot water it will, under the well-known properties of air, enormously expand. In actual practice it has been found possible to add, by reheating, one horse-power to each horse-power developed by compression, at one-eighth or

one-tenth the original cost of the latter. That is to say, if a given quantity of compressed air costs a dollar to generate, the further expenditure of ten cents in reheating will double its power to do work. Theoretically the total efficiency thus obtained is actually greater than if the same amount of coal had been burned in an ordinary steam-engine and the power thus generated used direct. In practical use it is slightly less.

With the reheating method came a marked improvement in compressing apparatus, in which advance, it is to be noted, the United States has taken the lead. There yet remained a third problem, the same that has baffled the electrician—the question of storage. It was found that air at high pressure is much more penetrating, for example, than steam. Now, however, storage reservoirs have been devised in which the loss from leakage is next to nothing. These are known as Mannesmann tubes, and are simply large, seamless, rolled flasks of mild steel, with but a single vent. These are of high tensile strength. and may be drawn to hold air at very high pressure. With these it is possible to reach a large amount of stored-up energy in a very small space, and without danger, for if these tubes were to burst they would not fly in pieces, but would simply rip, like a leather bag, and beyond a swift rush of air and a loud report there would be no hazard to life. With low cost of generation, economy of expenditure in use, and perfect storage, it remained only to develop the special appliances for its immediate application to give compressed air its present wide range. In particular did this advance make possible its use as a traction agent. The first to experiment in this field were the French. The surface cars to be seen in New York City are by no means the first of their kind. They represent simply an improvement of the well-known Mekarski system of Paris. The latter has been in use in the French capital for something like ten rears, and a number of different attempts have been made from time to time to introduce it in this country. The requirements of a service for the narrow and crowded thoroughfares of a city like New York, however, seemed too great for the French system to meet. It was not economical, it was heavy and clumsy, and not feasible in other ways. The Hardie system—that in use in New York City—seems to have overcome these difficulties, and, furthermore, to be able to cope with the peculiar and perplexing conditions which obtain in the congested portions of a large city where the overhead trolley is not allowed.

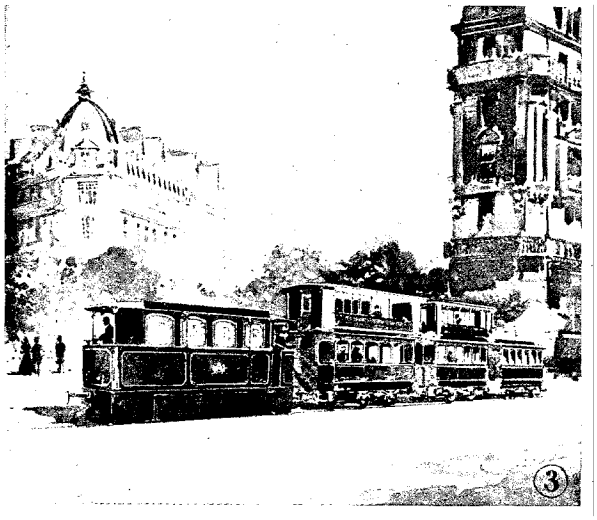

Fig. 3.—Motor and Train in Paris (Mékarski System).

The cars are identical in appearance, both inside and out, with the type used on the cable or trolley roads, and the motive apparatus is completely hidden from view. It is located entirely beneath the seats and the body of the car, and is controlled by a mechanism that out-wardly does not differ from that for the control of the trolley. As the car starts, there is some slight hissing noise, and that is all. The ease with which it is started and stopped does away with the exasperating jerkiness of the cable; and aside from this, the system seems much safer, from the quickness with which the car may be brought to a standstill. In cases of prime necessity the whole force of the compressed air which the car carries may be thrown upon the wheels, and a

dead stop from full speed effected in a few feet. The immediate operation would no doubt slightly dislocate the occupants of the car, but it would be a departure from the system of wholesale depopulation and dismemberment which seems the special mission of the trolley. Like the latter, the speed of the motor is variable within wide limits, and much faster runs may be made at night, after the theatre, than is possible with the cable.

The mechanical features of the Hardie motor are exceedingly simple. An apparatus substantially identical with that of the steam-locomotive is supplied with compressed air, instead of steam, from tubes extending under the car seats. These tubes or cylinders are charged at two thousand pounds pressure, but when used the air is passed through a reducing valve, bringing the pressure to one hundred and thirty pounds. The reheating is effected by forcing the air, before its entrance to the valves of the driving-gear, through a chest of superheated water. In its passage through the latter the air is not only enormously expanded, but is converted into a sort of vapor, which, it is claimed, gives a peculiarly high efficiency. In all about fifty-one cubic feet of air is stored in the steel reservoirs, affording sufficient power to run the car fifteen or eighteen miles, including stops. The time occupied in recharging the cars with air and hot water is not more than a minute, so that there is practically no delay. The motor mechanism itself consists of two simple. link-motion reciprocating engines, which do not differ materially from ordinary steam-engines in their features, save in their use of air. By ingenious device it is possible to start the motor at any point, and there is thus no chance for stoppage over a "dead-centre." It is to be noted that this ability to start promptly under all conditions is a feature the lack of which has been one of the principal causes of failure of air-motors heretofore tried. Still another unique device is that by which a surcharge of air is admitted to the valves in order to accelerate the start, this pressure being shut off when the car gains its full speed. The manipulation of the whole mechanism is exceedingly simple, requiring no more skill than that of the ordinary trolley. The cars move in either direction with equal facility, and are under such perfect control that it is possible to start and stop them within a space of two inches. The same lever that re leases the brakes operates the starting-gear, and, similarly, brakes are applied and the air cut off with a single motion of the wrist. As I am confining myself entirely to the favorable points of the new motor, it may be added that further claim is made for the Hardie motor that it is the cheapest of all systems in point of first cost, since it requires no subterranean or overhead construction, and can be introduced on any track. Each car being independent and carrying its own motive power, no derangement of machinery at the central station or of the "feeder"—that is, of cable or trolley—can affect the whole line. Electrical storms are, of course, no danger, and there are no cables to strand and to afford the reporters for the morning newspapers material for exciting tales of running-away cars. There is no dissipation of energy, as in the case of steam, trolley, or cable, since practically only such power is taken as is actually used. The smoothness of operation represents a further economy, alike of temper and profanity,

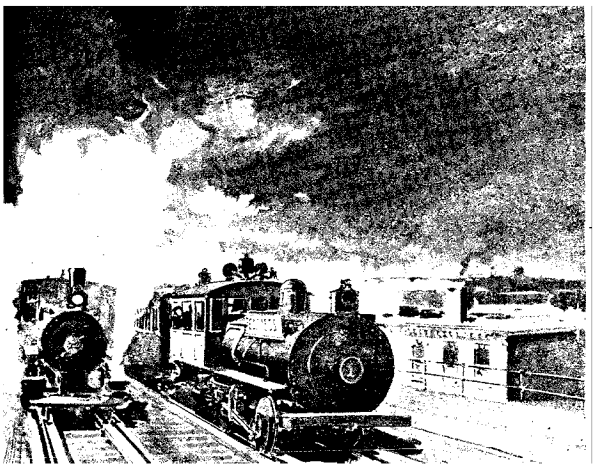

Fig. 4.—Compressed Air Locomotive to be operated on the Elevated Railroad in New York City.

so that I doubt not the ingenious promoter will hardly fail to represent them as an agency for the moral regeneration of the community.

There remains the single factor of economy of operation . Diligent inquiry upon this point failed to bring forth any really illuminative facts; probably the latter do not exist. It is claimed that compressed air for traction is less than half as expensive as the cable; and there is a further gain in fixed charges, since the cost of insulation is immensely less than in that of the cable of the underground trolley. This can only be demonstrated by actual experiment, such as is now being made in New York. Aside from the trial on the 125th Street line, the same company has now under construction a compressed-air locomotive which will be employed to draw a train of five cars on the Manhattan Elevated Railroad between Rector and 58th Streets. Still further, besides the Hardie system, the Metropolitan Railroad has on trial yet another compressed air motor, the invention of W. J. Knight and Joseph H. Hoadley. Three cars of the latter construction are now in use on the Lenox Avenue line. but of their capability, as of their special points of construction, the inventors do not as yet care to speak.

The detail of the development of compressed air here given is necessarily but a sketch, but even this will disclose that much more than a beginning has been made. The general use of the new power will come only with the advent of large central distributing stations, from which it can be had as freely as is gas or water now. While in a sense it is a rival of electricity, yet it is not impossible that it will soon become a yokemate rather. The alternating electrical current may be employed to transmit cheap power long distances, as from Niagara and the Pennsylvania culm-banks, and this power converted into compressed air at the point of consumption—electricity being probably the best agent for distance transmission, and compressed air the most mobile form of power for immediate use. Even this is mere conjecture, however, since there is at least one engineering genius—a man of remarkable achievement, at that—who has distinctly in view the compression of air at great water-powers like Niagara, conveyance by pipe-lines, at enormous pressure, so far as New York or Philadelphia, and delivery at prices with which electricity cannot compete. But whether as yokemates or rivals, it must be clear to the dullest imagination that present conditions are but a stage, and that we are but on a threshold of the day when these two forces, harnessed and trained, will, from their cheapness and availability, and in their infinite application, lift a considerable share of the burden of physical toil from the shoulders of the race.